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**COMBUSTION AND PERFORMANCE
CHARACTERISTICS OF METHANOL
PLUS HIGHER ALCOHOLS IN
A SPARK IGNITION ENGINE**

DEPOSITORY

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COMBUSTION AND PERFORMANCE CHARACTERISTICS OF METHANOL
PLUS HIGHER ALCOHOLS IN A SPARK IGNITION ENGINE
Final Report

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FOREWORD

The transportation fuel industry is experiencing a time of change. Depletion of fossil petroleum reserves and increased costs of petroleum-derived fuels are forcing the industry toward alternative fuels. There is also an urgent need to develop cleaner-burning fuels to relieve the pressures of air pollution on our environment.

The best current option for a future transportation fuel base appears to be methanol, a simple alcohol that can be abundantly produced from natural gas, coal or biomass. When synthesized from coal or biomass, the result is a mixture of methanol with small quantities of higher alcohols, such as ethanol, butanol and propanol. This methanol plus higher alcohols mixture (MPHA), in combination with gasoline and additional ethanol, is likely to become the transportation fuel of the future. With some modifications, existing automobile engines can use methanol more efficiently than gasoline and simultaneously release fewer emissions.

This project developed base-line data on the physical and combustion characteristics of MPHA in mixtures with gasoline. The data should prove invaluable to automotive engineers for designing more-efficient and cleaner operating engines capable of using wide ranges of fuel mixtures. ENR is pleased to have sponsored this research and hopes that the information provides a useful base for redefining our transportation energy resources.

Don Etchison, Director
Illinois Department of Energy
and Natural Resources

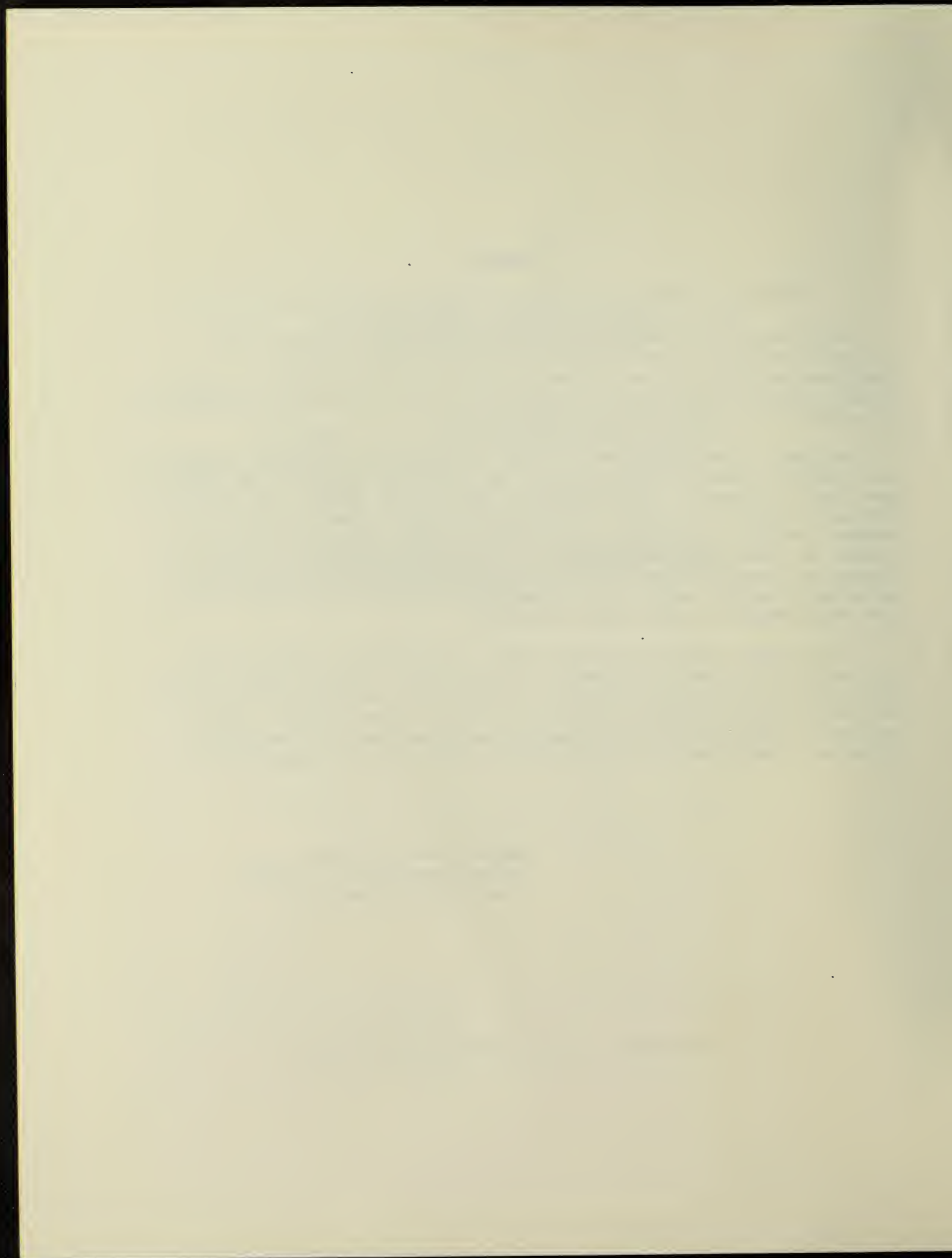
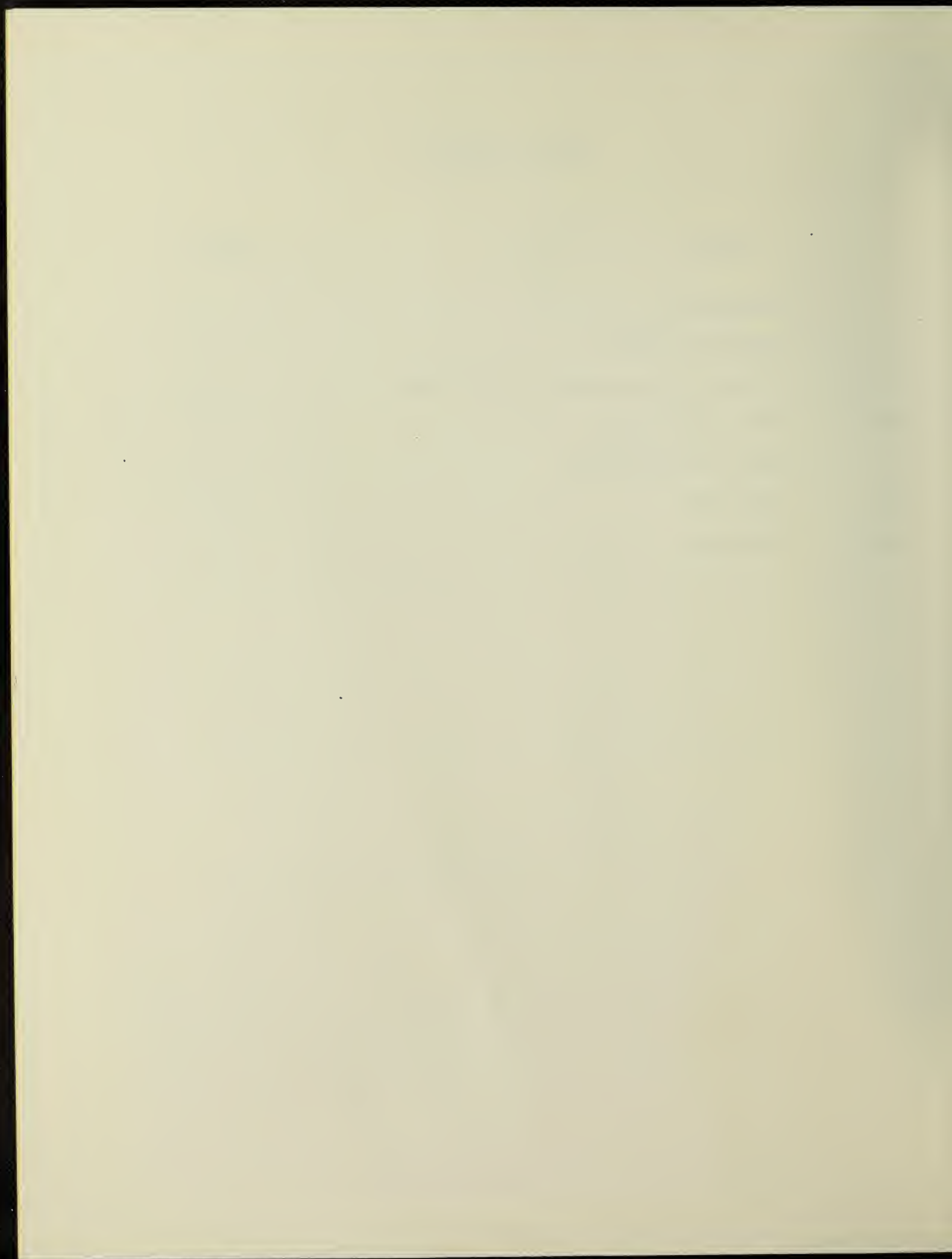


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CHAPTER I

INTRODUCTION

Methanol received wide attention as a fuel for the spark ignition engine in pure form or as a blend with gasoline. Methanol today can be produced from the nation's abundant resources, coal, waste products from farms and municipalities, as possible additional feedstock[1-10]*. It is probable that the synthetically produced methanol fuel of the future will, in fact, be a mixture of methanol and higher alcohols, rather than pure methanol. In the low pressure methanol-production process, the purity of methanol is achieved by conventional distillation methods. The other components are higher alcohols which can be extracted at lower levels of the distillation column [12]. The use of methanol plus higher alcohols instead of pure methanol reduces the production cost of the fuel, in general.

The composition of alcohol fuels derived from coal and other sources depends upon the manufacturing process, process operating conditions, and properties of the raw material [11]. The alcohol blend which is representative of the product which could be obtained from a methanol synthesis process using a non-selective catalyst contains by volume, 75% methanol, 5% ethanol, 7.5% 1-propanol, and 12.5% 2-methyl-1-propanol [13]. This composition of alcohols was used as the Methanol Plus Higher Alcohols fuel. Because of different compositions of indolene-methanol blends and indolene-MPHA blends,

* Numbers in brackets refer to references in Chapter VII.

the physical properties and performance parameters need to be examined at various operating conditions. The fuel specifications, are listed in Table 1.1. One of the most important properties of methanol and MPHA is the high octane number, which allows its use at higher compression ratios, resulting in an increase in the thermal efficiency of the spark ignition engine. The objective of this work is to investigate the physical and chemical properties of MPHA-indolene blends, and the combustion characteristics.

The experiments were performed at various methanol and MPHA concentrations (from 0 to 100%) in the blend. The compression ratio was varied from 5:1 to Knock Limited Compression Ratio (KLCR). The fuel properties, performance and combustion characteristics were also evaluated.

In Chapter II, the experimental facility has been described. In Chapters III, IV and V, the data analysis, theoretical analysis, and results are presented, along with a discussion. Chapter VI contains the conclusions and recommendations.

TABLE 1.1

Fuel Inspection Data

Property	Indolene	Methanol	Ethanol	N-Propanol	Isobutanol
Formula	---	CH ₃ OH	C ₂ H ₅ OH	C ₃ H ₇ OH	CH ₃ CH(CH ₃)CH ₂ OH
Specific gravity 60°F/60°F	0.75	0.793	0.8111	0.8045	0.8030
Boiling temp. °F	80-430	147	171.1	207	226.2
Flash Point °F	-45	54	68	81	95
Autoignition temp. °F	495	---	---	---	---
Vapor pressure at 68°F, MMHG	---	97.68	48.1	14.9	8
Flamability limits					
upper, %	7.6	36	---	13.7	10.6
lower, %	1.3	6.0	3.3	2.1	1.7
Vapor density (air = 1)	3	1.1	1.3	2.1	2.5

CHAPTER II

EXPERIMENTAL SETUP

Engine and Instrumentation

A standard single cylinder CFR spark ignition engine with variable compression ratio, manufactured by Waukesha Motor Company was used in these experiments. The combustion chamber, with a flat cylinder head, is shown in Fig.2.1. One minor modification was made to facilitate measurement of the flame speed. A spark plug with extended electrodes was placed in the magnetic pick-up plug position, and an ionization probe was placed in at the spark plug position. This was done in order to allow deep probing into the combustion chamber, without interfering with the piston motion. This engine was coupled to a General Electric CD-77 dynamometer, which can absorb 30 horsepower and deliver 25 horsepower. Fig.2.2 shows the overall layout of the experimental setup.

A Bendix breakerless ignition system is used in this engine. This system consists of an ignition coil, a control unit, a spark plug cable, and a spark plug. An AC spark plug with extended electrodes was cut to align it with the center line of the travelling probe, as in Fig.2.3

The intake air system consists of an air-filter, a laminar flow element (Meriam 50MW20-2), and an air heater. The absolute intake air pressure is measured at the intake of the laminar flow element, with an inlet absolute pressure gage (Meriam 310EF10). The air flow

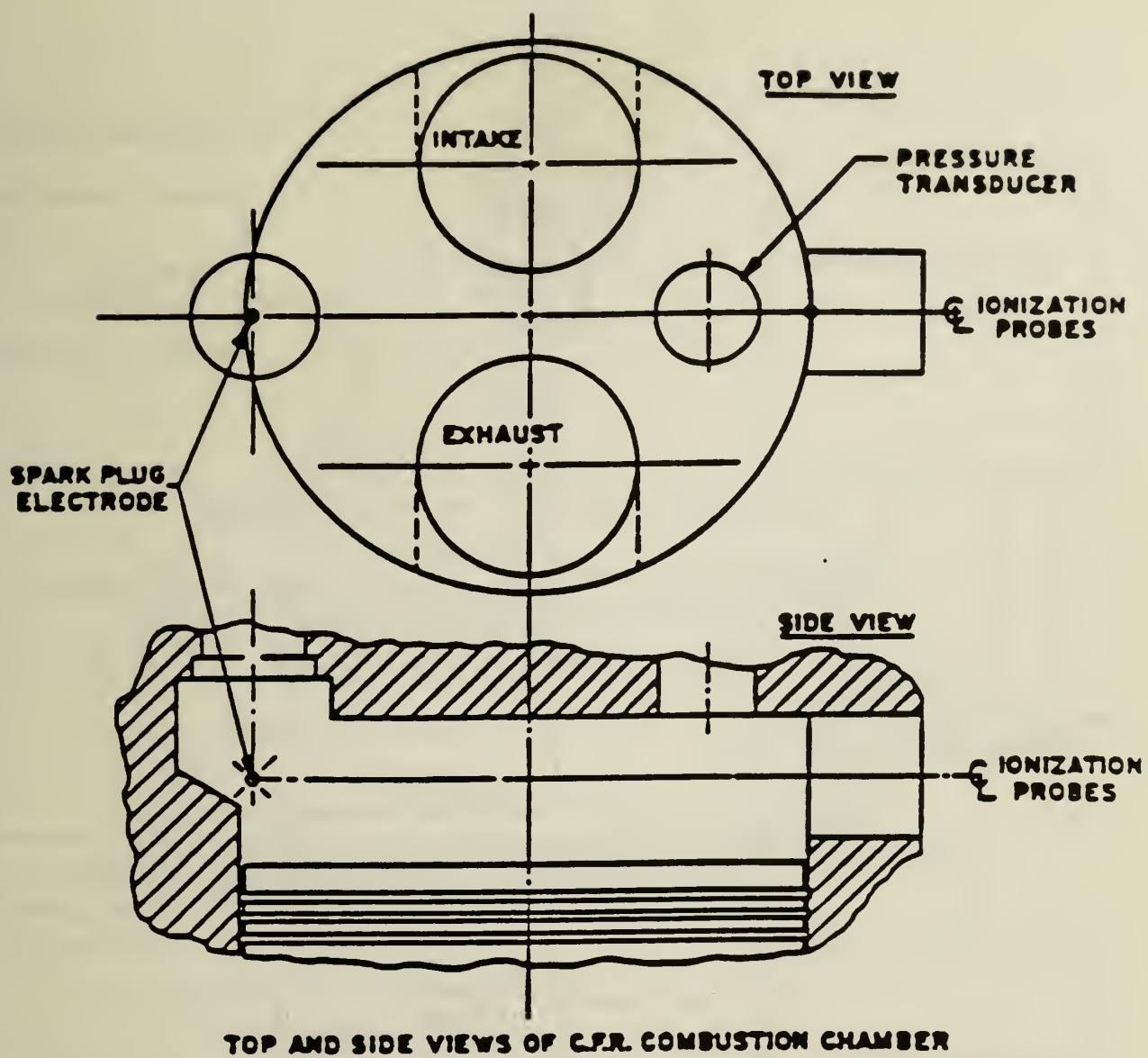


Fig. 2.1 Top and Side Views of C.F.R. Combustion Chamber

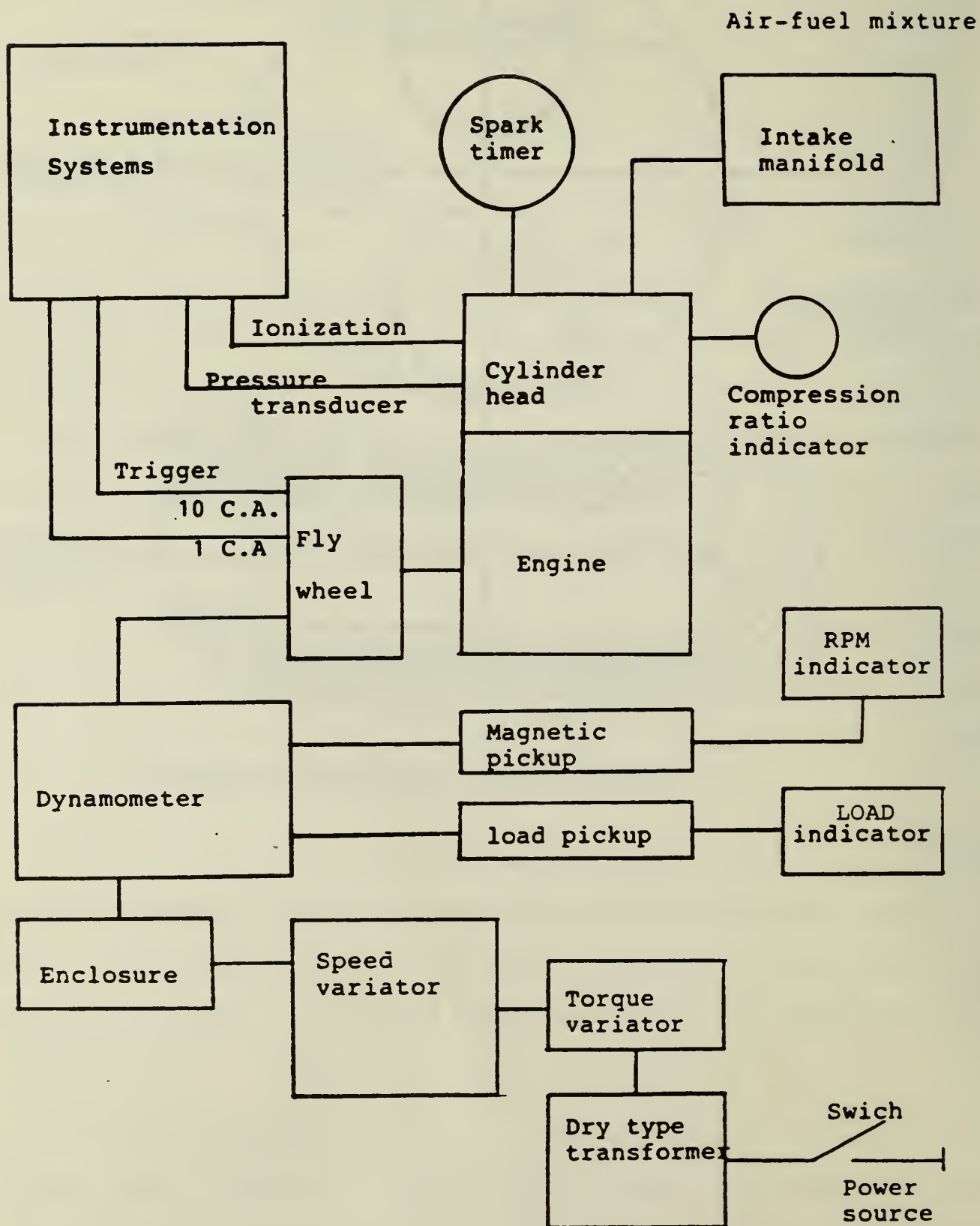


Fig.2.2 Overall engine and instrumentation systems.

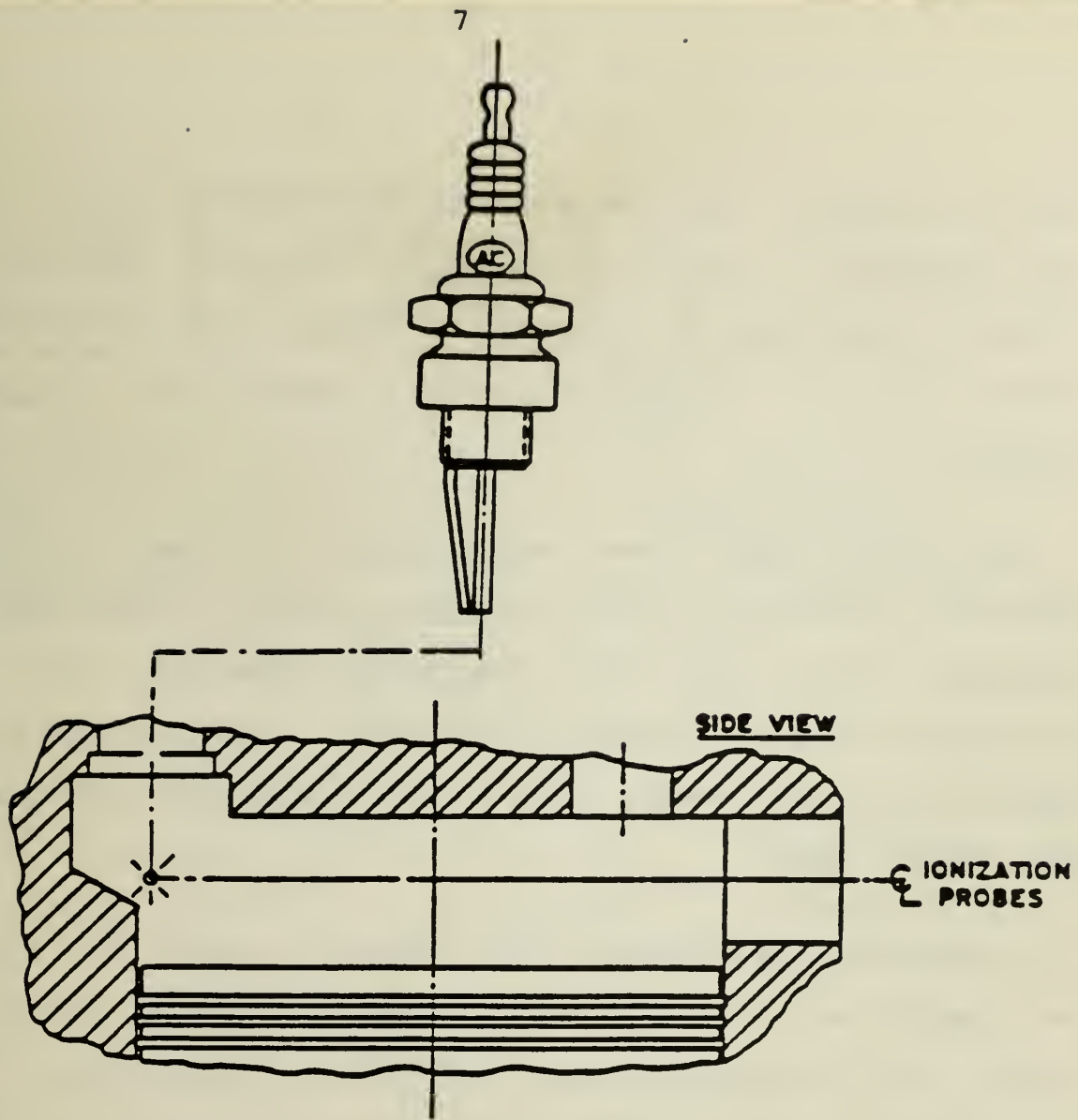


Fig. 2.3 Spark plug and its alignment with the probe

rate is computed from the pressure differential across the laminar flow element, with a differential pressure indicator (Meriam 40HE35). The inlet air temperature is kept constant at 120 F, by an electric heater. Fig.2.4 shows the layout of the fuel-air intake system.

The engine speed was measured by a shaft encoder (BEI-H250-60-A-7460R-EM16), and a frequency counter. The engine load was measured with a load cell (Interface SSMAJ250) and a digital transducer-indicator (Newport Q2000SGR3). The load cell is used to measure the engine torque and to find the minimum spark advance for best torque (MBT).

A high frequency response AVL pressure transducer type KQP300CVK was used to measure the cylinder pressure. This transducer is water cooled, and is vibration and temperature-compensated. It is used in conjunction with a Kistler 504E Charge Amplifier. Prior to measuring cylinder pressure, the transducer was coated with silicone, and calibrated along with its mating amplifier.

The fuel system consists of a fuel reservoir, a fuel pump, a fuel container, a weighing scale, an electronic timer, a fuel filter, and an injection nozzle. Clear indolene, methanol, ethanol, propanol, and butanol (industrial grade) were used in this study. The rate of fuel consumption is measured by a weighing scale and timer.

Flame propagation time is measured with the ionization probe. It consists of a water cooled probe, and an electronic amplifier-

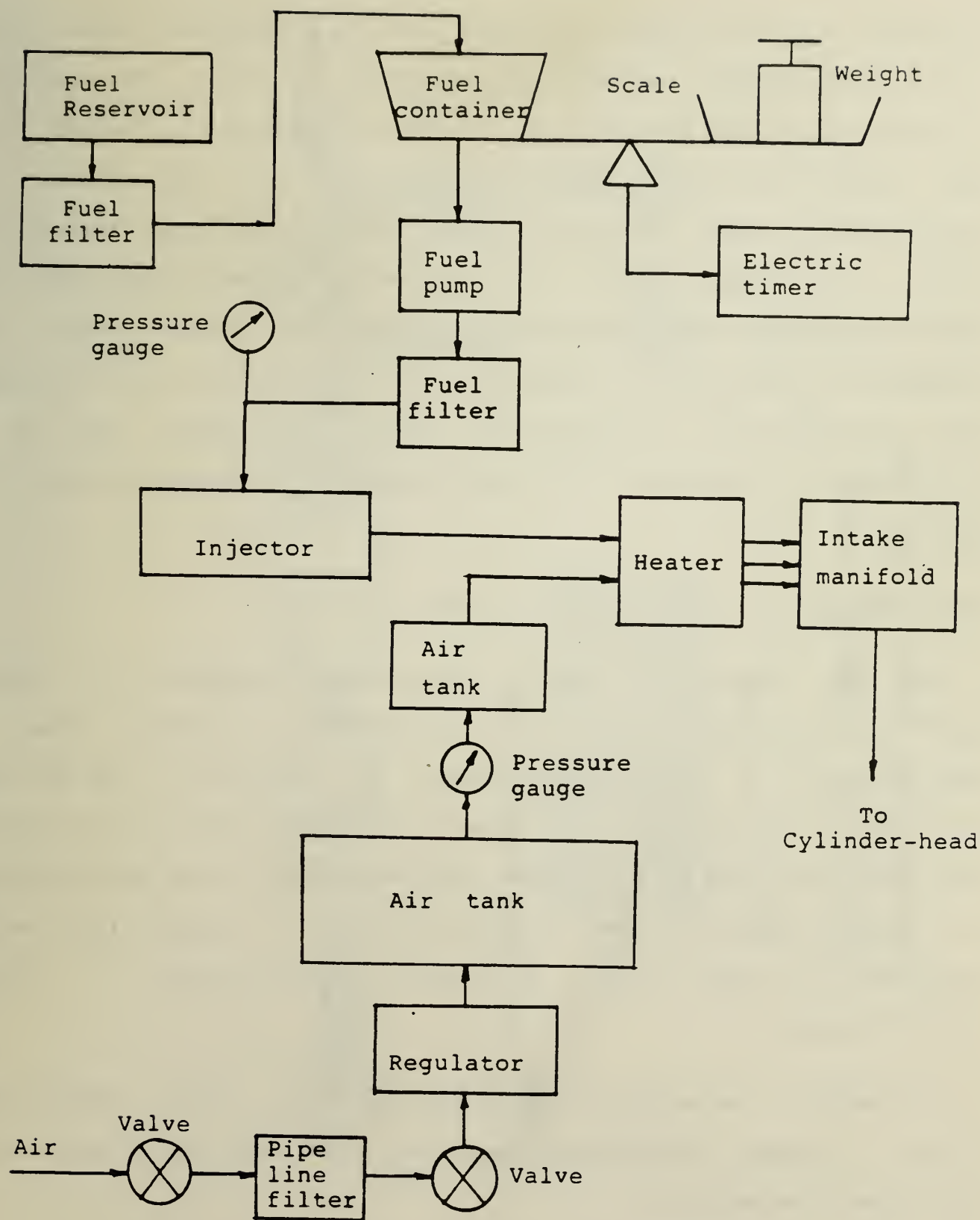


Fig. 2.4 Intake fuel-air system.

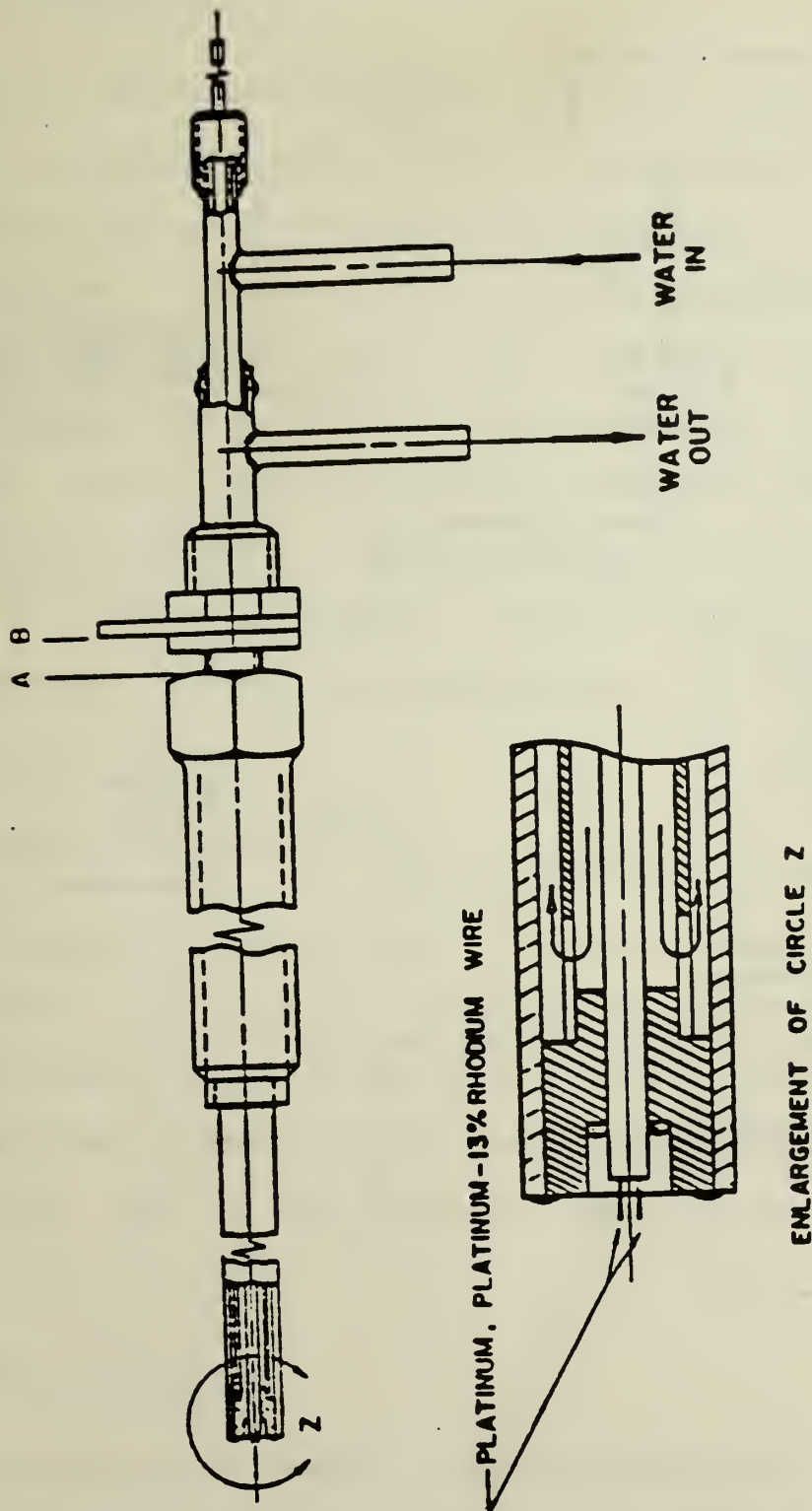
counter. Fig.2.5 shows the layout of the ionization probe. This probe was designed, and manufactured here, at U.I.C. The probe can traverse along its center line, inside the combustion chamber, and can be held at any desired distance from the spark plug. The electrodes of the ionization probe consist of two thermocouple wires- a platinum one and a platinum-13%rhodium wire. They are separated by MgO insulation, and enclosed in an inconel sheath. The diameter of the wires is 0.063 in. The outer ends of the electrodes are connected to the electronic amplifier-counter, and to an oscilloscope, the signal from which is input to the data acquisition system.

Data acquisition and recording system

The data acquisition and recording system consists of 1 CAD at 2000 rpm acquisition rate, 2 A to D channels, 1 interval timer, 1 TDC detector, 1 parallel data latch, an IBM XT PC, and an FX85 printer. Fig.2.6 shows the layout of the data acquisition system. The following data were recorded and monitored on the oscilloscope:

1. Instantaneous cylinder pressure signal.
2. Spark signal which indicates the spark timing.
3. Ionization probe signal.
4. Shaft encoder signal.

In addition, mass flow rate of air and fuel, torque, temperatures of oil, coolant, and exhaust gas were also recorded to obtain the performance parameters.



WATER-COOLED TRAVELING PROBE
DEVELOPED BY WAYNE STATE UNIV.

Fig. 2.5 Water cooled traveling probe

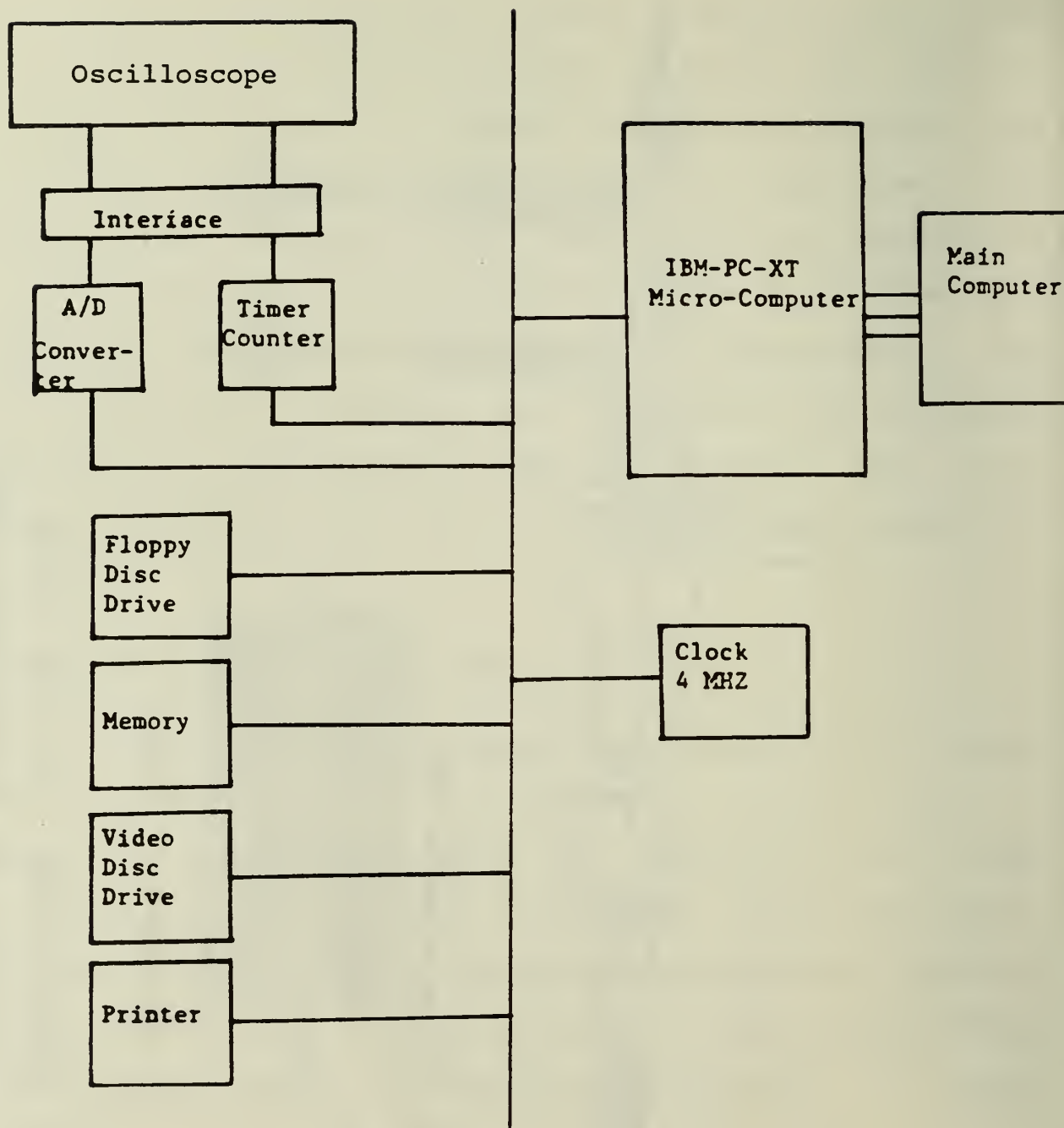


Fig. 2.6 Data Processing System

Equipment Preparation

The engine was given a complete overhaul, including a cleaning of the piston, cylinder head, intake and exhaust valves, intake and exhaust manifolds, cylinder pressure transducer, injection pump and cooling system. The pressure transducer was calibrated and mounted flush with the cylinder head. The load cell-indicator system was also calibrated. The intake and exhaust valves were carefully lapped to ensure proper seating before installation in the engine. Calculations were made to determine the stoichiometric air to fuel ratios for all the fuel blends, as described in the appendix. The data recording sheets were developed as seen in the sample calculation.

The experiments were divided into two parts.

The first part was to obtain the Minimum spark advance for Best Torque (MBT), and the Knock Limited Compression ratio (KLCR) with indolene-methanol and indolene-MPHA blends. The second part was to obtain detailed combustion and performance characteristics of all fuel blends. The engine operating conditions and test matrix are given in Table 2.1-2.3.

Experimental Procedure

The following experimental procedure was used:

1. Prior to running experiments, the engine and other equipments were prepared as stated in the previous section.
2. The checklist shown in the appendix was followed to motor and fire the engine.

TABLE 2.1

TEST MATRIX FOR INDOLENE-METHANOL BLENDS

CR	100%I	70%I+ 30%M	50%I+ 50%M	30%I+ 70%M	100%M
5.0	101	104	108	112	116
6.0	102	105	109	113	117
7.0	103	106	110	114	118
8.0		107			119
9.0					120
10.0			111	115	121

TABLE 2.2

TEST MATRIX FOR INDOLENE-MPHA BLENDS

CR	100%I	70%I+ 30%MPHA	50%I+ 50%MPHA	30%I+ 70%MPHA	100%MPHA
5.0	101	122	126	130	134
6.0	102	123	127	131	135
7.0	103	124	128	132	136
8.0		125			
9.0					
10.0			129	133	

Table 2.3

GENERAL OPERATING CONDITIONS

Spark plug gap (in.)	0.045
Spark timing	MBT
Valve clearance (in.)	0.008 + 0.001 (hot)
Oil (crankcase)	Synthetic oil design for methanol
Intake air temperature (°F)	120 $\pm$ 2
Equivalence ratio	1 + 0.02
Speed (RPM)	1000 $\pm$ 5
Compression ratio	5 to KLCR
Fuel	indolene, indolene- alcohols blends.
Injection pressure (psi) (manifold)	1200
Intake air pressure	WOT

3. The compression ratio was adjusted and varied to obtain the KLCR.
4. The engine speed was set at a constant value of about 1000 rpm.
5. The equivalence ratio was set to about 1.0 by using the calculation sheet shown in the appendix.
6. The engine was run for a period of an hour, at the operating conditions given in Table 2.3 to ensure steady state operation before recording the data.
7. In the first part, the spark advance was changed from 20 deg.BTDC to 50 deg.BTDC in increments of one degree and the load cell reading was recorded. The spark advance at which the load cell indicated maximum torque is called the MBT spark advance.
8. The compression ratio was varied from 5:1 to KLCR in increments of 1.

Performance Parameters

The parameters which have been studied in this work are; 1. brake power, 2. brake specific fuel consumption and 3. brake thermal efficiency. The tests started with pure indolene . Methanol was then added by 30%, 50% and 70%. Tables 2.1 and 2.2 show the experimental test matrix. The compression ratio was varied from 5:1 to KLCR, i.e., when engine knock became audible. The knock was observed from the pressure trace.

Burning velocity

The parameters which have been studied in this part of the study are: 1.Flame speed, and 2.Burning velocity. The volume fraction of methanol and MPHA were varied by 0.1, 0.2, 0.3, and 0.4. The compression ratio was kept constant at 5:1. The flame propagation time was measured at eight locations during flame propagation.

CHAPTER III: DATA ANALYSIS AND SAMPLE CALCULATION

In this chapter, the data analysis and a sample of the calculation will be presented. The data analysis consists of two parts: 1. Flame travel time analysis. 2. Pressure-Time history analysis.

Flame Travel Time Analysis

The flame travelling time is taken as the time interval between the spark timing and the arrival of the flame at the tip of the ionization probe. A high speed time counter is used to measure the flame travelling time in milliseconds. The spark and ionization signals are also monitored on the oscilloscope. For each run, 512 data points are temporarily stored in the microcomputer memory and later transferred to files.

After determining the 512 data points for the flame propagation time for each position of the ionization probe, an arithmetic average is performed on the data by the program TIMEAVG. This program also checks and compares the mean with all the data points, excludes bad data points, and recalculates the average flame travel time (Bad data points were defined as above 1.5 times or below 0.5 times the mean).

This process was repeated for all the different probe positions between the spark plug and the chamber wall. The time average data

were then curve fitted using a fifth degree polynomial to give the distance vs. time relationship for the flame travel process. In order to improve the accuracy of the curve fitting routine, additional points were inserted near the end points.

The apparent flame speed is computed from the slope of the distance-time curve by a computer program APPBVEL. This program calculates the apparent flame speed at every degree crank angle by using the polynomial coefficients as input.

Pressure-Time History Analysis

The cycle-to-cycle variation in pressure traces is a common problem in spark ignition engines. The combustion characteristics calculated from the model are very much dependant upon the pressure, and its derivatives. An attempt has been made in the literature to use the average pressure of many cycles. Pattersons[18] has recommended that 2000 pressure cycles be used for a representative average cycle, and Peter[19] has recommended 300 cycles. Recently Mattavi et al [20] used 96 consecutive cycles to obtain the average pressure data. Following was the procedure used in this study to obtain the average pressure data. The high speed data acquisition system was used to collect 80 consecutive cycles for each run. The pressure was recorded at every $1/2$ deg. crank angle. The arithmetic average of these cycles was taken to obtain one average pressure cycle.

Sample Calculation

Flame Speed

The average time of 512 points for eight different probe positions are shown below:

Flame Travel Time	Distance	Improved Flame Travel Time
0.85000E-02	8.49	
0.80000E-02	8.50	
0.75000E-02	8.25	
0.72186E-02	8.09	0.72186E-02
0.62738E-02	7.09	0.64068E-02
0.56755E-02	6.09	0.56755E-02
0.49691E-02	5.09	0.49691E-02
0.42186E-02	4.09	0.43507E-02
0.36305E-02	3.09	0.36305E-02
0.28760E-02	2.09	0.28760E-02
0.22500E-02	1.50	
0.17868E-02	1.09	0.17934E-02
0.10377E-02	0.69	0.10979E-02
0.07500E-02	0.50	
0.05000E-02	0.25	
0.02500E-02	0.15	
0.00000E-02	0.00	

Fig. 3.1 is a plot of the distance vs. flame travel time. The circular symbols are the inserted data points to increase the accuracy of the polynomial curve fitting. After curve fitting, the coefficients are used to calculate the flame speed at every crank angle. The appendix contains plots of flame speed vs. crank angle calculated from APPBVEL program.

Mass Fraction Burned

As shown in Fig. 3.2, ignition delay and combustion interval are calculated from the mass fraction vs. degree crank angle curve. These data are obtained from the computer simulation program SPMIX.

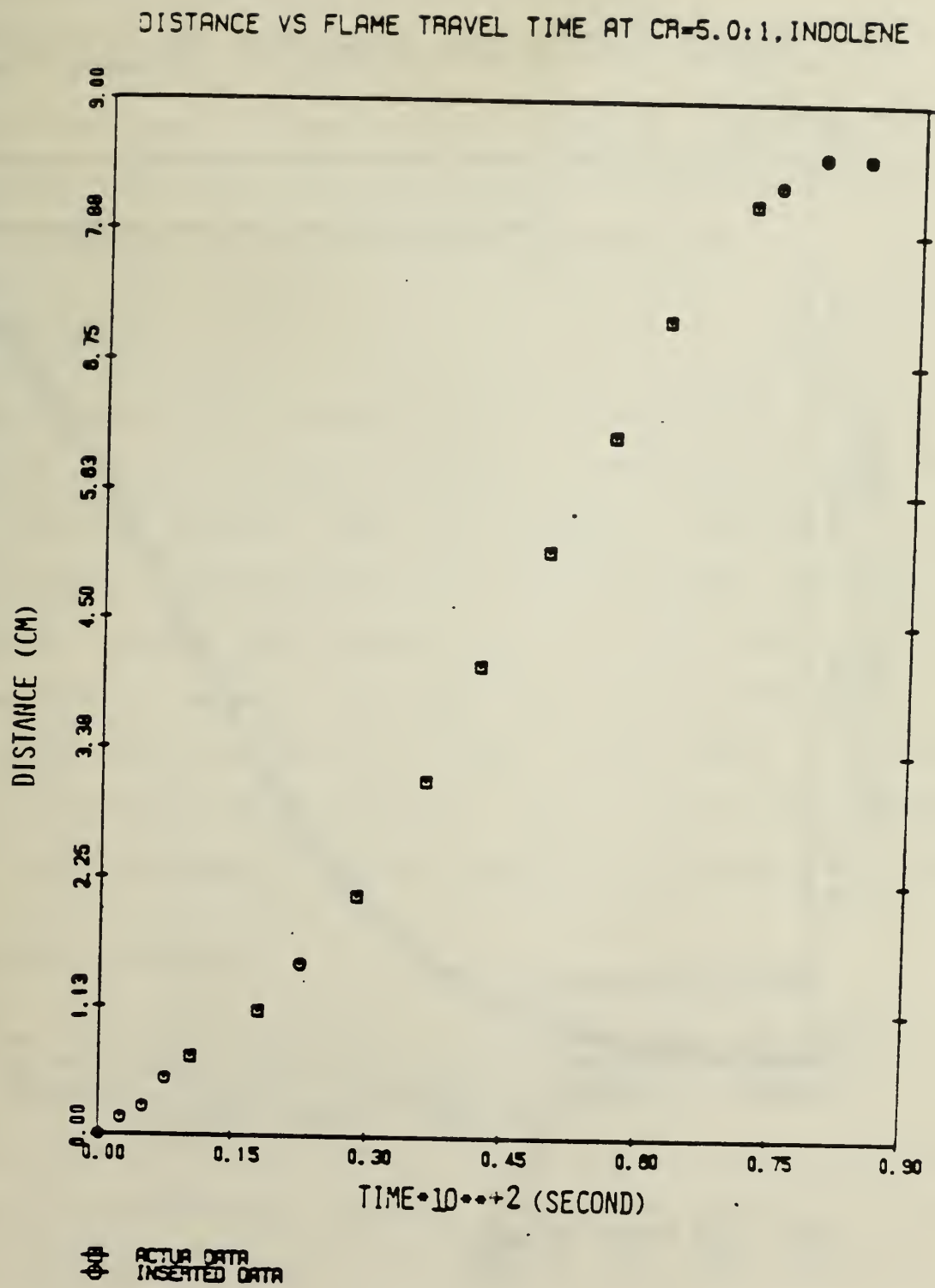


Fig. 3.1 Distance-Time curve

MASS FRACTION BURNED VS DEGREE CA

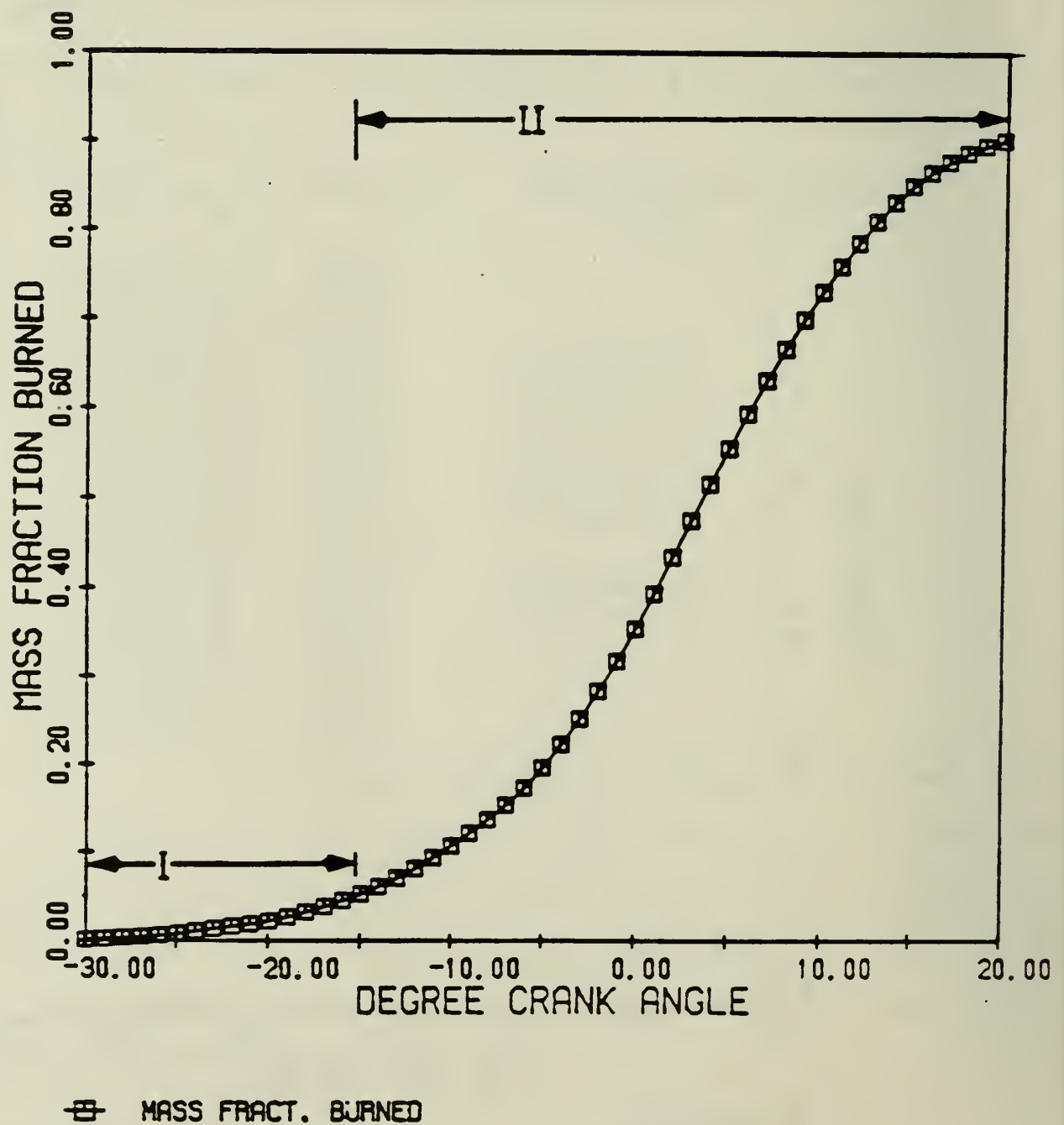


Fig. 3.2 Mass fraction burned vs °CA

Determination of Combustion Characteristics

The fundamental quantities from which combustion can be characterized are, actual burning velocity, ignition delay, and combustion interval. The following is an outline of the procedure used in calculating these fundamental quantities.

Ignition Delay

The period of time from spark to the point when heat release is detected, is called ignition delay. This period can be calculated from the p-t diagram where the pressure increases drastically. Alternatively, this period can be obtained from the mass fraction burned vs. degree crank angle curve. Ignition delay can be defined as the interval between the spark and the time when 5% of the mass is burned. This definition is used in this study to calculate the ignition delay period. From Fig. 3.2, it can be seen clearly that after 5% of the mass is burned, the rate of burning rises sharply.

Combustion Interval

After the ignition delay period, the fully developed turbulent flame propagates across the combustion chamber. Combustion interval is the period of time from the end of the ignition delay period to the end of combustion. Keeping in mind the accuracy of the model, the combustion interval is defined as the time between 5% to 90% mass burned. During this interval, the flame propagates with a finite thickness through the unburned mixture. Fig. 3.2 shows the definition of combustion interval from the mass fraction burned vs.

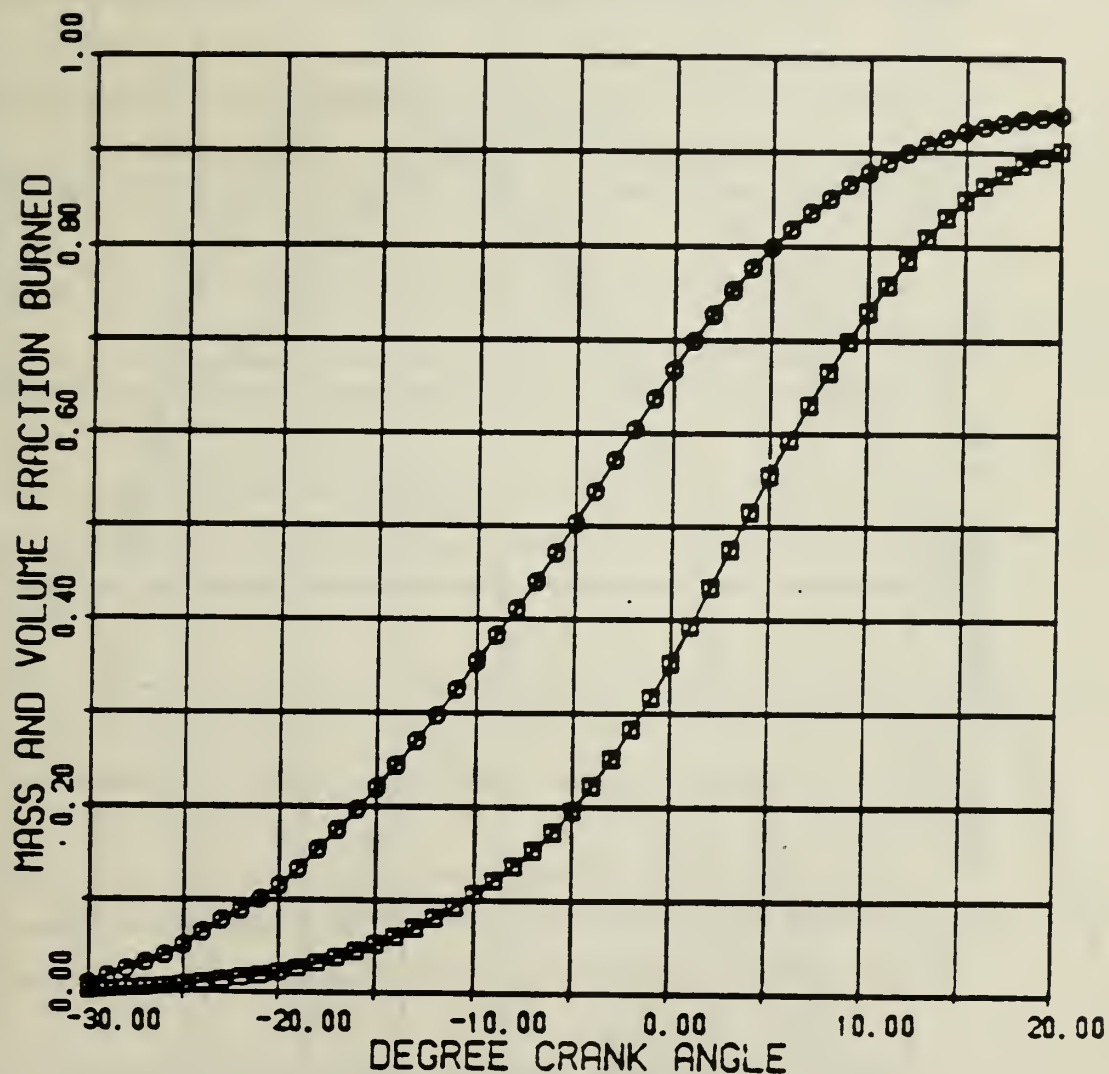
degree crank angle curve.

Burning Velocity

The burning velocity can be calculated from the flame speed and the pressure-time data. the flame front area at every crank angle is needed for these calculations. Flame front area was calculated from the spherical flame propagation model. Fig. 3.4 is a plot of the flame front area vs. flame radius.

Fig. 3.5 shows the curves of flame speed, expansion velocity, and actual burning velocity as functions of crank angle degree, for indolene.

MASS AND VOL. FRACT. BURNED VS DEGREE CA



○ MASS FRACT. BURNED
 □ VOL. FRACT. BURNED

Fig. 3.3 Plot of mass and volume fraction burned vs degree crank angle at base conditions.

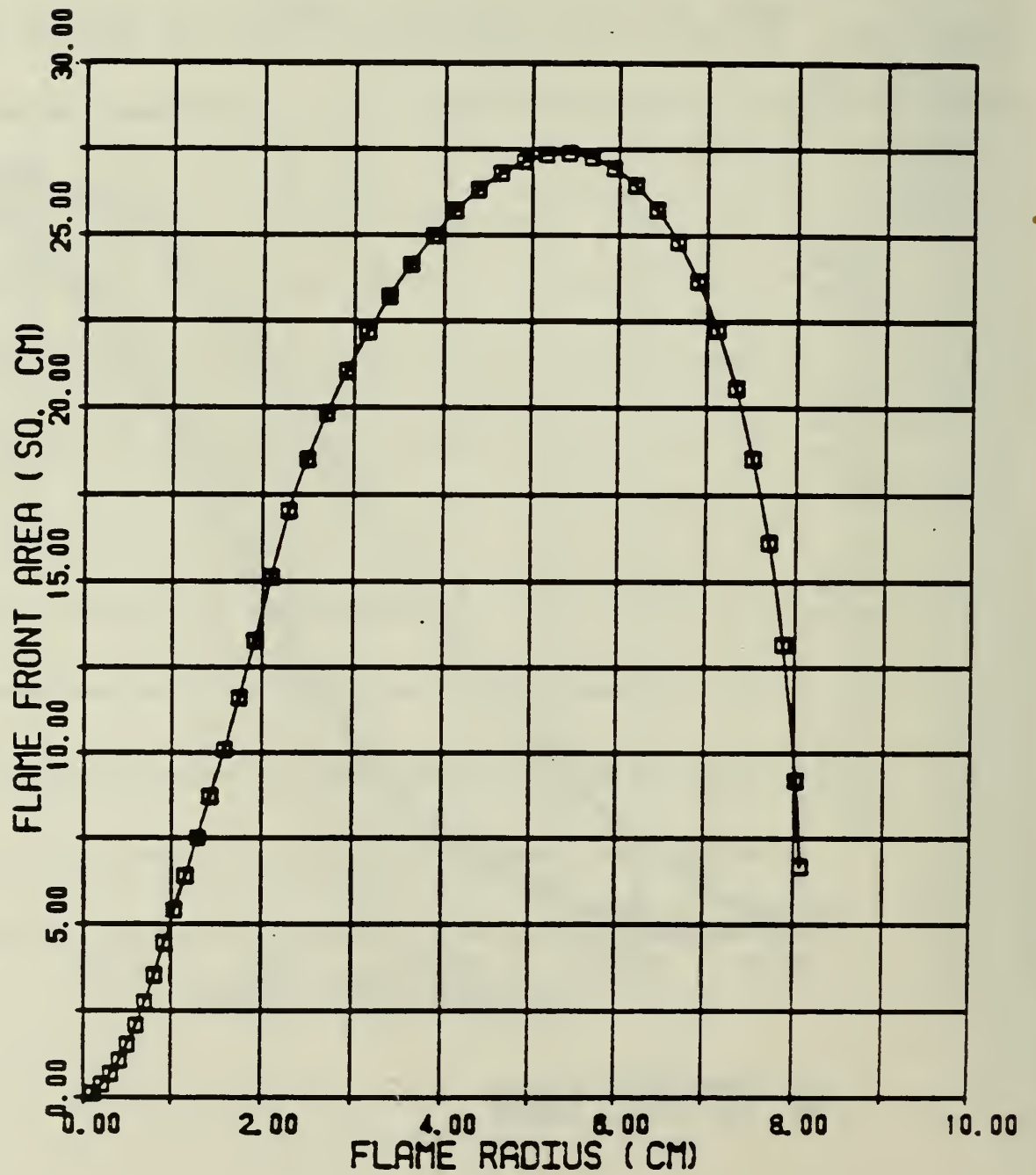
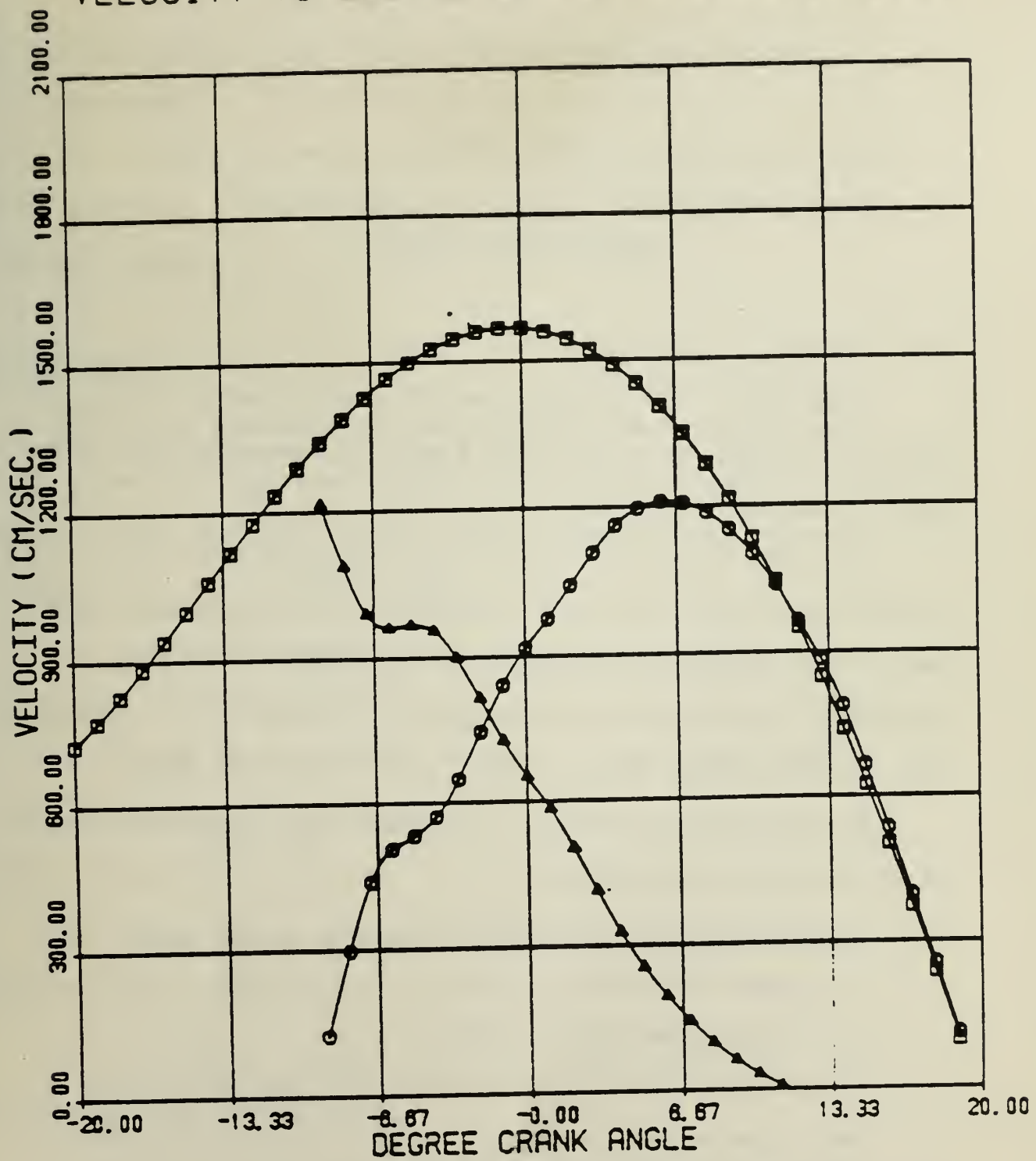
FLAME RADIUS VS SPHERICAL FLAME FRONT AREA AT $Ca=5$ 

Fig. 3.4 Flame front area vs flame radius

VELOCITY VS DEGREE CA FOR INDOLINE, CR=5



□ APP. BURNING VEL.
 ○ ACTUAL BURNING VEL.
 ▲ EXPANSION VELOCITY

Fig. 3.5 Burning Velocity vs °CA

CHAPTER IV

THEORETICAL ANALYSIS

4.1 Models of Combustion

Modeling of the combustion phenomena inside the spark ignition engine is relatively simple if compared with modeling of diesel combustion.

In spark ignition engine, the mixture of air and fuel is pre-vaporized and can be assumed as an homogeneous mixture. The combustion process is treated as a constant pressure combustion and immediately followed by an isothermal mixing between newly burned and the burned gases.

As proposed by Bracco, the engine combustion models can be classified as follows:

1. Zero-dimensional models
2. Quasi-dimensional models
3. Multi-dimensional models

The above models are discussed earlier in Chapter 2.

Zero-dimensional model with the chosen three zones control volume is employed in this analysis because it gives fairly good results and is relatively simple to use.

4.2 Derivation of the Energy Related burning Velocity

Equation

The burning velocity equation is derived by two analytical methods: Thermodynamics analysis and combustion mechanism analysis.

Assumptions:

1. During combustion, the cylinder contents are made up of three distinctive regions, one of burned part, one of unburned part and the flame (reactive region) in between separates the two regions above.

2. The mass burning rate of the fuel air mixture is due to the chemical reaction only and is equal to the rate of mass gained to the control volume of the burned part.

3. Pressure surrounding the control volume of burned region is uniform throughout the combustion process.

4. There are no effect of the potential and kinetic energy in the combustion.

5. At initial stage, there is a very small (or none) quantity of the burned mass.

6. The burned gases consist of a uniform mixture of products of combustion.

7. The unburned mixture is made up of a homogeneous mixture of air, fuel vapor and residual products of combustion.

8. The systems behave according to ideal gas law.

9. Heat transfer is confined to convective exchange between the gasses and the walls.

10. The energy required to propagate the flame from initial to final position is denoted as the flame work and is defined as:

$$W_{cv} = W_{exp} - W_{comp} - W_{loop} + W_f \quad (4.1)$$

Where

W_{cv} = Total net work for a given system

W_{exp} = Expansion work

W_{comp} = Compression work

W_{loop} = Loop work

W_f = Flame work

4.2.1 Thermodynamics Analysis

Fig.4.1 illustrates the initial and final states of an arbitrary chosen control volume of the burned gases. Fig. 4.2 shows the system after the combustion of a charge element δm and the uniform pressure surrounding the control volume.

The conservation of mass, together with First law of thermodynamics is applied to the system as follows.

* Conservation of Mass

At any instant during the process, the continuity

equation is written for the system as:

$$\int_0^t \left(\frac{dm}{dt} \right)_{cv} dt + \int_0^t \sum \dot{m}_e dt - \int_0^t \sum \dot{m}_i dt \quad (4.2)$$

For a period of time t , equation (4.2) becomes

$$(\dot{m}_2 - \dot{m}_1)_{cv} + \sum \dot{m}_e - \sum \dot{m}_i = 0 \quad (4.3)$$

where

$\dot{m}_1, \dot{m}_2$ = Mass rate of the burned gases at initial stage and final stage respectively.

$\dot{m}_i, \dot{m}_e$ = Mass burning rate enter to and exit from the control volume respectively.

From the assumptions 2 and 5,

$$\dot{m}_1 = 0$$

$$\dot{m}_e = 0$$

after arranging and let $\dot{m}_b$ denotes the mass burning rate, equation (4.3) would become:

$$\sum \dot{m}_2 = \dot{m}_i = \dot{m}_b \quad (4.4)$$

* First Law of Thermodynamics

Writting the First law for the uniform state, uniform flow (USUF) process, at any instant of time for the control volume of the burned part, during the process, we have:

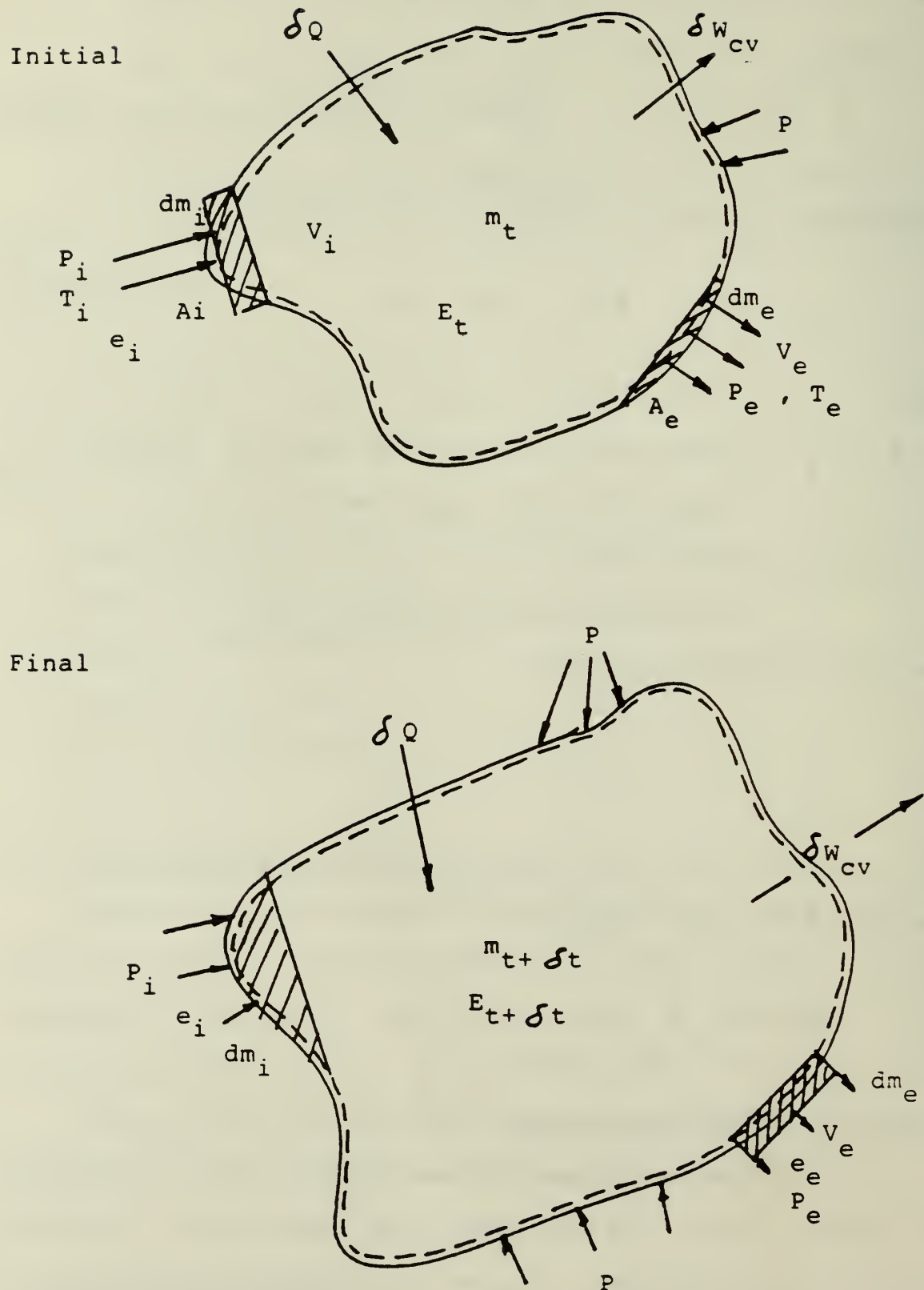


Figure 4.1: Initial and final state of the system.

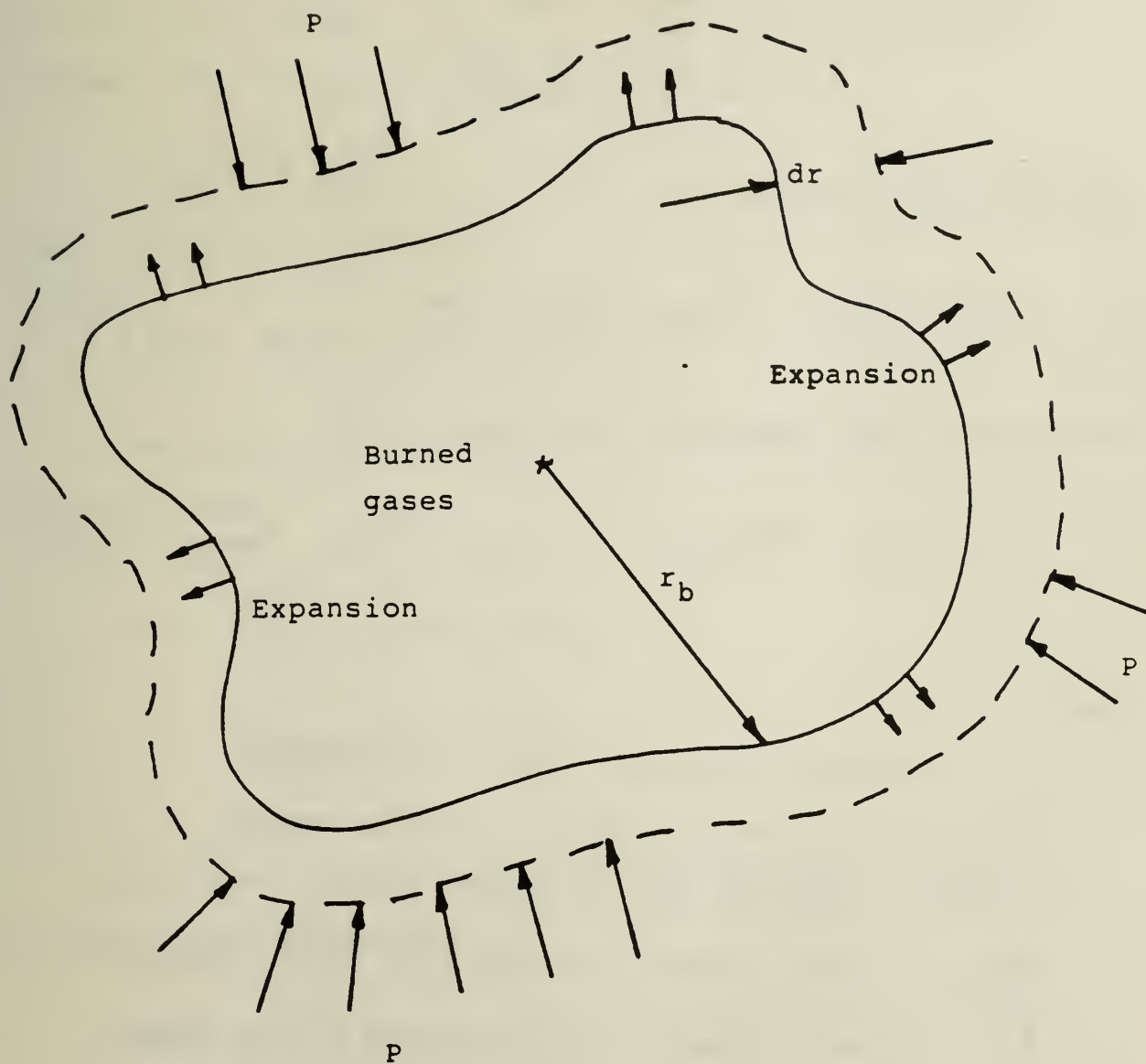


Figure 4.2: Uniform pressure surrounding the system.

$$\dot{Q}_{cv} + \sum \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) = \left(\frac{dE}{dt} \right)_{cv} + \dot{W}_{cv} + \sum \dot{m}_e \left(h_e + \frac{v_e^2}{2} + gz_e \right) \quad (4.5)$$

Defferentiate equation (4.1)

$$\dot{W}_{cv} = \dot{W}_{exp} - \dot{W}_{comp} - \dot{W}_{loop} + \dot{W}_f \quad (4.6)$$

Substitute (4.6) into (4.5) and solve for $\dot{W}_f$

$$\dot{W}_f = \dot{Q}_{cv} + \sum \dot{m}_i h_i - \left(\frac{dE}{dt} \right)_{cv} - \dot{W}_{exp} + \dot{W}_{comp} + \dot{W}_{loop} \quad (4.7)$$

where

$\dot{W}_f$ = Rate of flame work done by the system.

$\dot{Q}_{cv}$ = Rate of heat transfer in to the system.

$\sum \dot{m}_i h_i$ = Rate of total enthalpy input.

$\left(\frac{dE}{dt} \right)_{cv}$ = Rate of energy accumulated in the system .

$\dot{W}_{exp}$ = Rate of expansion work done by the system.

$\dot{W}_{comp}$ = Rate of compression work done on the system.

$\dot{W}_{loop}$ = Rate of loop work.

Let dE_T/dt denotes the right hand side of equation (4.7)

and multiply both side by dt:

$$\oint W_f = m_b w_f dt = dE_T \quad (4.8)$$

From Fig.4.2, pressure surrounding the control volume is uniform, the work done is:

$$\begin{aligned}
 w_f &= p(dv_b) = (p)(A)(dr_b) = (F/A)(A)(dr_b) \\
 &= F(dr_b)
 \end{aligned}
 \tag{4.9}$$

At $t = 0$; $r_{b0} = 0$

$$t = t, dr_b = r_{bt} - r_{b0} = r_{bt}$$

$$w_f = F(r_{bt}) \tag{4.10}$$

Substitute (4.10) into (4.8)

$$\oint w_f = dE_T = \dot{m}_b (F)(r_{bt}) dt \tag{4.11}$$

From definition of momentum, per Kg mass of burned gases, we have:

$$\begin{aligned}
 F &= \left(\frac{dm_b}{dt} \right) (\text{velocity}) / dm_b \\
 &= \frac{\text{velocity}}{dt} = \frac{S_u}{dt}
 \end{aligned}
 \tag{4.12}$$

Substitute (4.12) into (4.11) integrating from zero to time t and rearranging:

$$S_u = E_t / (\dot{m}_b)(r_b) \tag{4.13}$$

$$S_u = w_f / (\dot{m}_b)(r_b)$$

Equation (4.13) represents the burning velocity as a function of the flame work, mass burning rate and the burned part.

This equation will be evaluated term by term and the

$$V_{f1} = V_{f0} + dV_{f1} = A_f (r_1 - r_0)$$

mass burning rate:

$$\dot{m}_{f1} = \dot{m}_{f0} + d\dot{m}_{f1} \quad (4.14)$$

Apply the First law to the flame element, neglecting the potential and kinetic energy, we have:

$$\dot{Q}_{cv} + \sum \dot{m}_i h_i = \frac{dE_{cv}}{dt} + \sum \dot{m}_e h_e + \dot{W}_{cv} \quad (4.15)$$

$$\text{From } \dot{m}_e = 0$$

and $\dot{W}_{cv} = \dot{W}_{exp} - \dot{W}_{comp} - \dot{W}_{loop} + \dot{W}_f$
equation (4.15), after solving for $\dot{W}_f$, would become,

$$\begin{aligned} \dot{W}_f = & \dot{Q}_{cv} + \sum \dot{m}_{f1} h_{u0} - dE_{cv}/dt - \dot{W}_{exp} \\ & + \dot{W}_{comp} + \dot{W}_{loop} \end{aligned} \quad (4.16)$$

Let dE_t/dt denotes the right hand side of equation 4.16)
and from definition of the flame work,

$$W_f = P_{f1} (dV_{f1}) = W_{f1}$$

$$W_{f1} = P_{f1} (A_{f1}) (r_1 - r_0) = F_f (r_1 - r_0) \quad (4.17)$$

and

$$\begin{aligned} \dot{W}_{f1} &= \frac{\delta W_{f1}}{\delta t} \\ &= \frac{F_f (r_1 - r_0)}{dt} \end{aligned} \quad (4.18)$$

Burning velocity is defined as the ratio of:

computation of the burning velocity will be given in sample calculation in Appendix 3.

4.2.2 Combustion Mechanism Analysis

Combustion process in the S.I. engine is assumed to progress in a stepwise manner as proposed by Patterson. It is described below with the assumption of no mass loss due to the movement of exhaust valve, i.e. total mass contained in the cylinder remains constant, during the process.

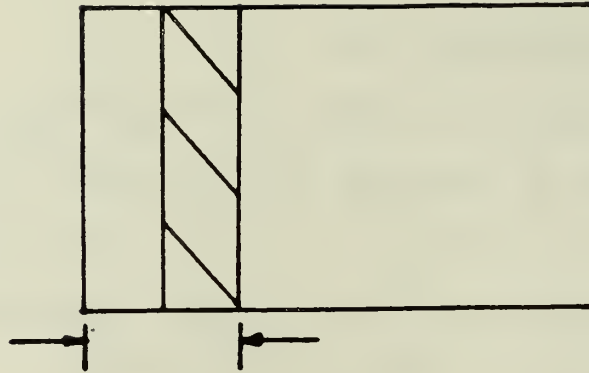
The whole system consists of three distinctive regions, one of burned part, one of unburned part and the flame between separates the two regions above.

The combustion is considered to take place in the intermediate processes as follows:

Process:

- 0-1 Constant pressure burning of an increment of the charge of thickness $dr = r_1 - r_0$.
- 1-2 Isothermal mixing between newly burned and burned gases.
- 2-3 Isentropic expansion of the burned gases.
- 3-4 Constant volume heat transfer.
- 4-5 Polytropic change due to piston motion.

Table I summarizes the parameters of the whole system at every stage of the process.

Initial

After the intake and isentropic compression processes in the cycle, the whole system is assumed to have a negligible quantity of the burned mass. Initial thermodynamic properties of the subsystems are given as:

Burned

Mass	$m_{b0} = 0$
Volume	$V_{b0} = 0$
Temperature	Not applicable
Pressure	N.A

Flame

Mass	$m_{f0} = 0$
Volume	$V_{f0} = 0$
Temperature	Not applicable
Pressure	N.A

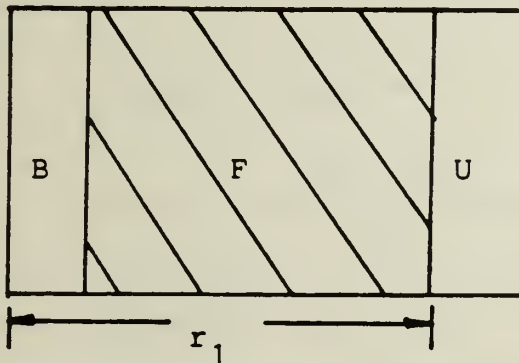
Unburned

Mass	m_{u0}
Volume	V_{u0}
Temperature	T_{u0}

Pressure

 P_{u0} Step 1: Constant Pressure Burning

An increment of mass, thickness of $dr = r_1 - r_0$, is burned at constant pressure. The flame temperature would be at its adiabatic flame temperature. Flame element will increase its volume, thus compressing the unburned part. The burned gases (if any) state is assumed remain unchanged.



The whole system is illustrated in the above figure.

Let's consider each of the subsystem as follows:

Burned

State remained unchanged.

Flame

The flame properties are changing from the initial condition to the end of this step and will be determined by applying the First law to the flame element. Properties at this stage are:

Pressure: P_{f1}

Volume:

$$S_u = \frac{dr}{dt} = \frac{r_1 - r_0}{dt} \quad (4.19)$$

Substitute (4.19) into (4.18) and solve for S_u , we have,

$$S_u = W_{f1} / F_f \quad (4.20)$$

$$\text{from } F_f = m_{f1}(a_{f1}) \quad (4.21)$$

and acceleration of the flame, a_{f1} , is defined as:

$$\begin{aligned} a_{f1} &= (\text{Velocity at 1} - \text{velocity at 0})/dt \\ &= \frac{(r_1 - r_0)}{dt} - \frac{(r_0 - r_0)}{dt} \end{aligned} \quad (4.22)$$

At initial condition, the flame radius $r_0 = 0$.

Substitute (4.22) into (4.21) then (4.20) and noting that

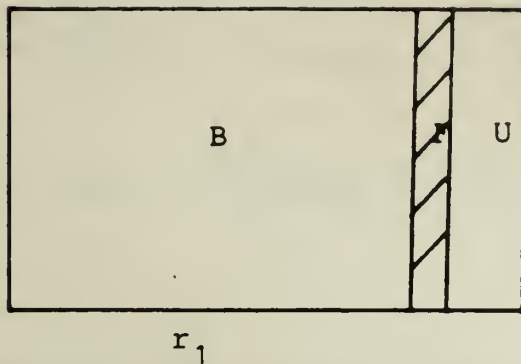
$$\dot{m}_{f1} = m_{f1} / dt$$

we would have:

$$S_u = W_f / (\dot{m}_{f1})(r_1) \quad (4.23)$$

The flame work, W_f will be determined by the equation (4.16) by the followed processes.

Step 2: Isothermal Mixing Between Newly Burned and Unburned Gases.



Burned Gases

The volume and pressure of this system is changing, the final volume is:

$$V_{b2} = V_{b1} + V_{f1} = V_{f1} \quad (4.24)$$

Temperature of the burned gases remained constant and is equal to the adiabatic flame temperature.

$$T_{b2} = T_{f1} = T_{adia}$$

The mass of the system is equal to the mass entering the burned gas. Since the initial mass $m_{b0} = m_{b1} = 0$ was assumed. The mass of the system after the mixing process is,

$$m_{b2} = m_{f1} = m_{u0} - m_{u1} \quad (4.25)$$

Applying the equation of state for an ideal gas, other properties of burned system can be calculated.

Flame

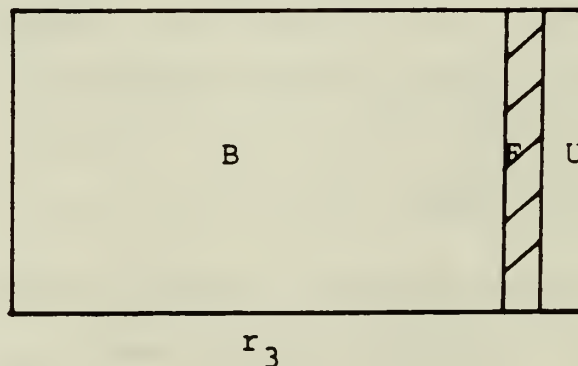
After the isothermal mixing process, the flame region

is reduced in its size with a negligible thickness and is assumed to be stationary and non reactive, until the end of the whole process, then it can start the cycle again.

Unburned Mixture

The state of this system remained unchanged, since it is assumed that the mixing process occurred between the burned and newly burned systems only.

Step 3: Isentropic Expansion of the Burned Gases.



Burned Gases: Expansion

The burned system undergoes through an isentropic expansion process. The burned mass remained constant. With the pressure rapidly increased from the previous process, the burned gases would compress the remaining unburned mixture and causing the piston to move downward.

Properties of burned gases can be calculated as follow:

$$\frac{T_{b3}}{T_{b2}} = \left(\frac{P_{b3}}{P_{b2}} \right)^{(K_p - 1)/K_p} \quad (4.26)$$

$$\frac{T_{b3}}{T_{b2}} = \left(\frac{v_{b2}}{v_{b3}} \right)^{(K_p - 1)} \quad (4.27)$$

Where

$$K_p = \frac{\bar{C}_{pp}}{\bar{C}_{vp}} = \frac{\sum x_i C_{pi}}{\bar{C}_{pp} - \bar{R}} \quad (4.28)$$

The expansion work can be determined by applying the First law:

$$\begin{aligned} W_{exp} &= U_{b23} \\ &= U_{b2}(T_{b2}) - U_{b3}(T_{b3}) \end{aligned} \quad (4.29)$$

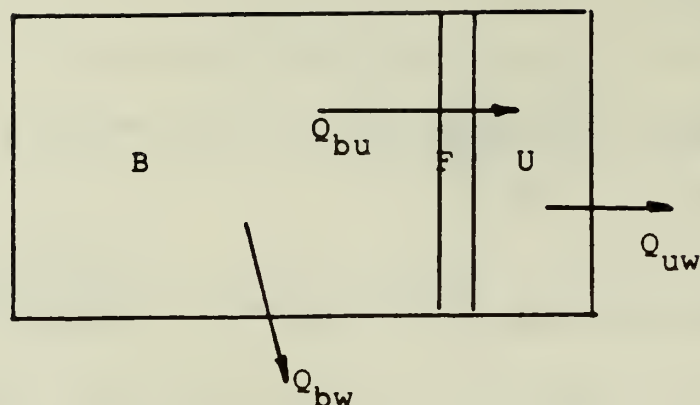
Unburned Mixture

The left-over unburned mixture, in contrast, would be compressed by the burned gas. The unburned mass is remained constant; by similiar application of the First law and equation of state for an ideal mixture, the properties of this system can be calculated.

With the assumption of the combustion process is completed, then there is a very little amount of unburned mixture presented.

Step 4: Constant Volume Heat Transfer

The overall heat transfer from the burned to unburned and to the walls, plus heat loss from unburned to the wall can be computed using Woschni's correlation. Because the



amount of heat transferred to the unburned mixture is just off-set by that from the unburned gas to the walls, as proposed by Patterson. The heat transferred from burned to unburned therefore can be ignored, the convective heat transfer can be calculated as follow:

$$\dot{Q}_{ui} = h_u(A_{ui})(T_{ui} - T_w)$$

$$\dot{Q}_u = \sum \dot{Q}_{ui} \quad (4.30)$$

and

$$\dot{Q}_{bi} = h_b(A_{bi})(T_{bi} - T_w)$$

$$\dot{Q}_b = \sum \dot{Q}_{bi} \quad (4.31)$$

Convective heat transfer coefficients are computed from Woschni's correlation and is given as:

$$h = 110 \left(\frac{P^{0.8}}{d^{0.2}(T^{0.53})} \right) \left[C_1 C_m + C_2 \frac{V_s T_1}{P_1 V_1} (P - P_0) \right] \quad (4.32)$$

The meaning of each term in this equation is given in Appendix 2.

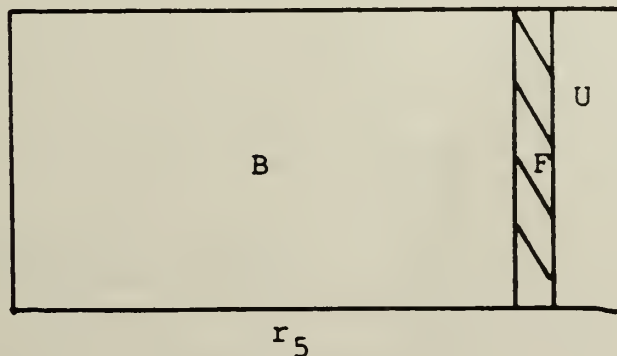
The wall area which exposed to the burned gas can be calculated as given in the below equation:

$$A = (\pi) \text{ Bore } \left[\frac{\text{Stroke}}{2} (1 + \cos\theta) + \text{Rod} \left(1 - \sqrt{1 - \frac{\sin^2\theta \text{ Stroke}^2}{4 \text{ Rod}^2}} \right) \right] \quad (4.33)$$

The Heat transfer area for burned gases is calculated from the spherical flame front model. Area for the unburned gases is the difference between total area (of chamber walls, piston and head) and the area in contact with the burned gases.

The wall temperature is measured with a thermocouple, this temperature is assumed to be the same everywhere on the wall surface temperature variations are quite small compared to the total temperature difference between the burned gases and the surfaces.

Step 5: Polytropic Change Due to Piston Motion



This final process is due to the piston motion, where-

ther it is polytropic expansion or compression, the following relation ship can be applied to the burned and unburned systems:

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4} \right)^{k-1}$$

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4} \right)^{\frac{k-1}{k}} \quad (4.34)$$

After this process is completed, the new cycle in the cylinder would take place with the same steps.

CHAPTER V

RESULTS AND DISCUSSION

This chapter describes the results obtained from the theoretical and experimental work.

Physical and Chemical Properties

When evaluating an alternative fuel, knowledge of its physical and chemical properties is essential for a thorough understanding of how the proposed fuel should perform in an I.C. engine. While there are some properties which are generally attributed only to either S.I. or C.I. engine fuels, many properties are common to both, as well as to turbine engine fuels. These properties include density, specific gravity, absolute viscosity, surface tension, distillation temperatures, flash point, and heating value. The fuel properties which have the greatest effect on S.I. engine operation are the distillation characteristics, flash point, heating value, and composition. In the present work, the following properties were measured or computed: 1. Distillation characteristic, 2. Heating value, 3. Flash point, 4. Water tolerance, 5. API gravity, and 6. Stoichiometric air-to-fuel ratio. Most of the properties were measured using standard ASTM procedure and equipment.

Distillation Characteristics

The distillation temperatures for the 15 fuels in this study are tabulated in Table 5.1. The data for indolene, indolene-methanol,

Fuel	IBP	5	10	20	30	40	50	60	70	80	90	95	E.P.	REC	RES.+ REC.	Loss
■	146	148	148	148	148	148	149	149	149	149	149	149	150	99.5	99.5	0.5
10 I + 90 ■	140	146	147	148	149	149	149	150	150	150	150	152	344	98.5	99	1.0
20 I + 80 ■	128	134	142	147	148	149	149	150	150	151	152	153	352	98	98.5	1.5
40 I + 60 ■	116	132	138	142	145	146	148	149	150	151	152	342	378	98.5	99.5	0.5
60 I + 40 ■	116	124	132	137	140	144	146	147	148	264	324	354	380	97	98	2.0
80 I + 20 ■	119	130	134	137	142	148	186	224	252	292	332	342	380	98	99	1.0
I	116	166	184	201	210	222	236	252	274	312	343	362	398	98	99.5	0.5
10 I + 90 MPHA	143	150	151	154	155	156	156	158	162	169	201	222	280	98.5	99	1.0
20 I + 80 MPHA	134	144	148	152	154	156	158	160	166	194	250	348	375	98	99	1.0
40 I + 60 MPHA	116	137	142	147	150	153	156	160	165	177	216	236	364	98	99	1.0
60 I + 40 MPHA	116	130	134	142	147	150	154	161	191	240	328	354	382	98	99	1.0
80 I + 20 MPHA	113	128	134	140	147	160	204	234	258	302	340	368	400	97.5	98.5	1.5
100 MPHA	152	154	154	155	156	157	158	160	164	172	200	220	240	99.5	100	0.0

and indolene-MPHA at the same blend levels are plotted in Figs.5.1-5.5. A substantial variation in distillation characteristics is evident. Fig.5.1 shows the distillation curves of pure indolene, methanol and MPHA. Because of a blend of many different hydrocarbons present in indolene, it shows a broad range of boiling points. Since methanol is a single component fuel, it has a constant boiling point temperature, and therefore, does not produce a distillation curve.

MPHA, on the other hand, is a blend of different alcohols, and produces a curve similar to that of indolene. It contains 75% by volume methanol, and a remaining 25% of other alcohols. Therefore, the distillation curve is straight line for about 75% distilled and then varies with temperature. The shape of the distillation curve will change as the composition of MPHA. For example, if extra ethanol or other higher is added to MPHA before mixing with indolene, the distillation curve will be different from that shown here.

Figs.5.2-5.5 show the distillation characteristics of indolene, indolene-methanol and indolene-MPHA fuels. The entire distillation curve is affected by the presence of alcohols. The constant temperature boiling due to methanol, which occurs slightly below 150 °F is shown clearly in all figures. At all blend levels, the indolene-MPHA blend is closer to indolene than the indolene-methanol blend.

Although distillation is a continuous process, only a few points

DISTILLATION CHARACTERISTICS INDO., METH. & MPHA

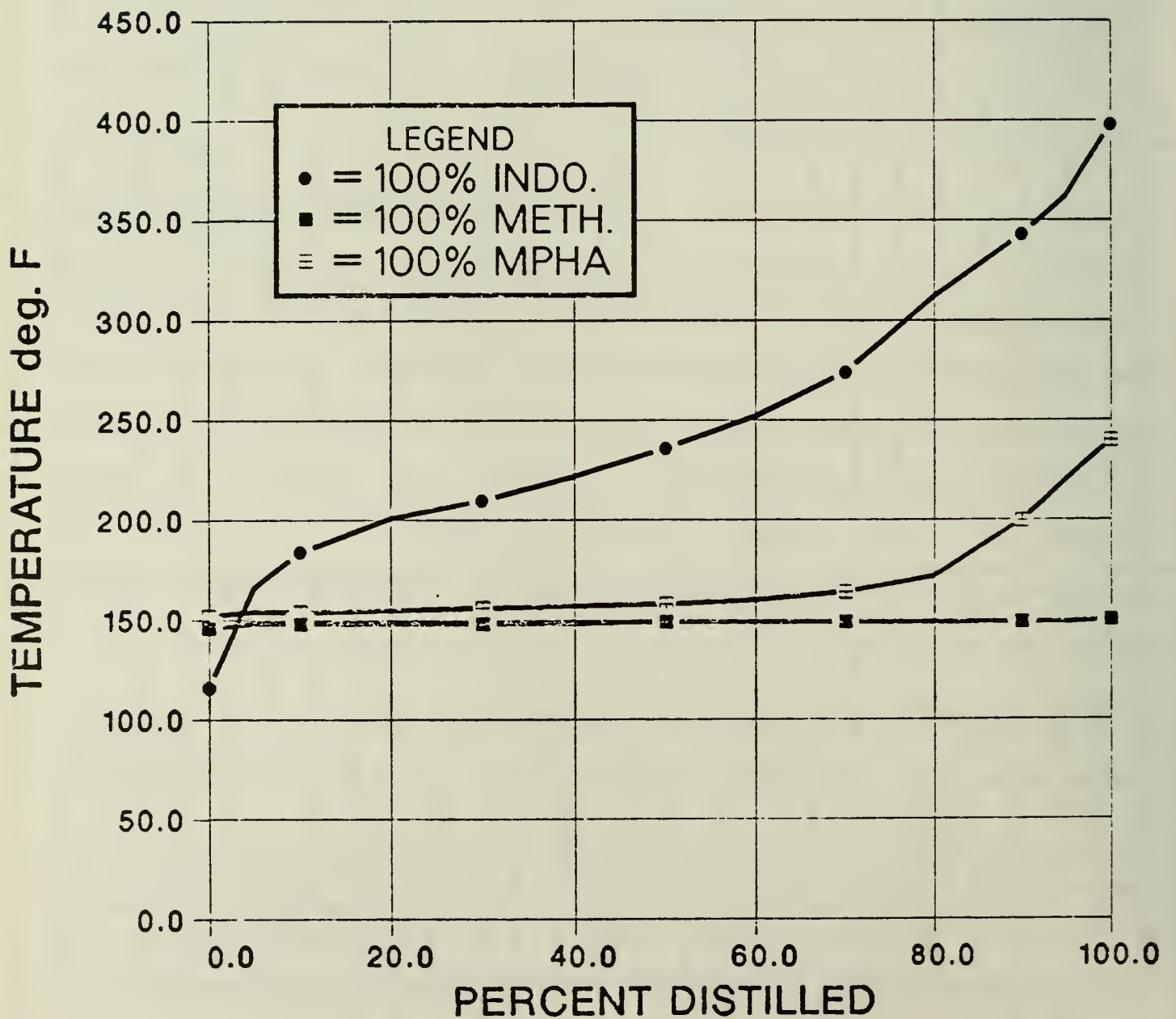


Fig. 5.1 Distillation Properties of Pure Fuels

DISTILLATION CHARACTERISTICS
INDO.-METH. & INDO.-MPHA BLENDS

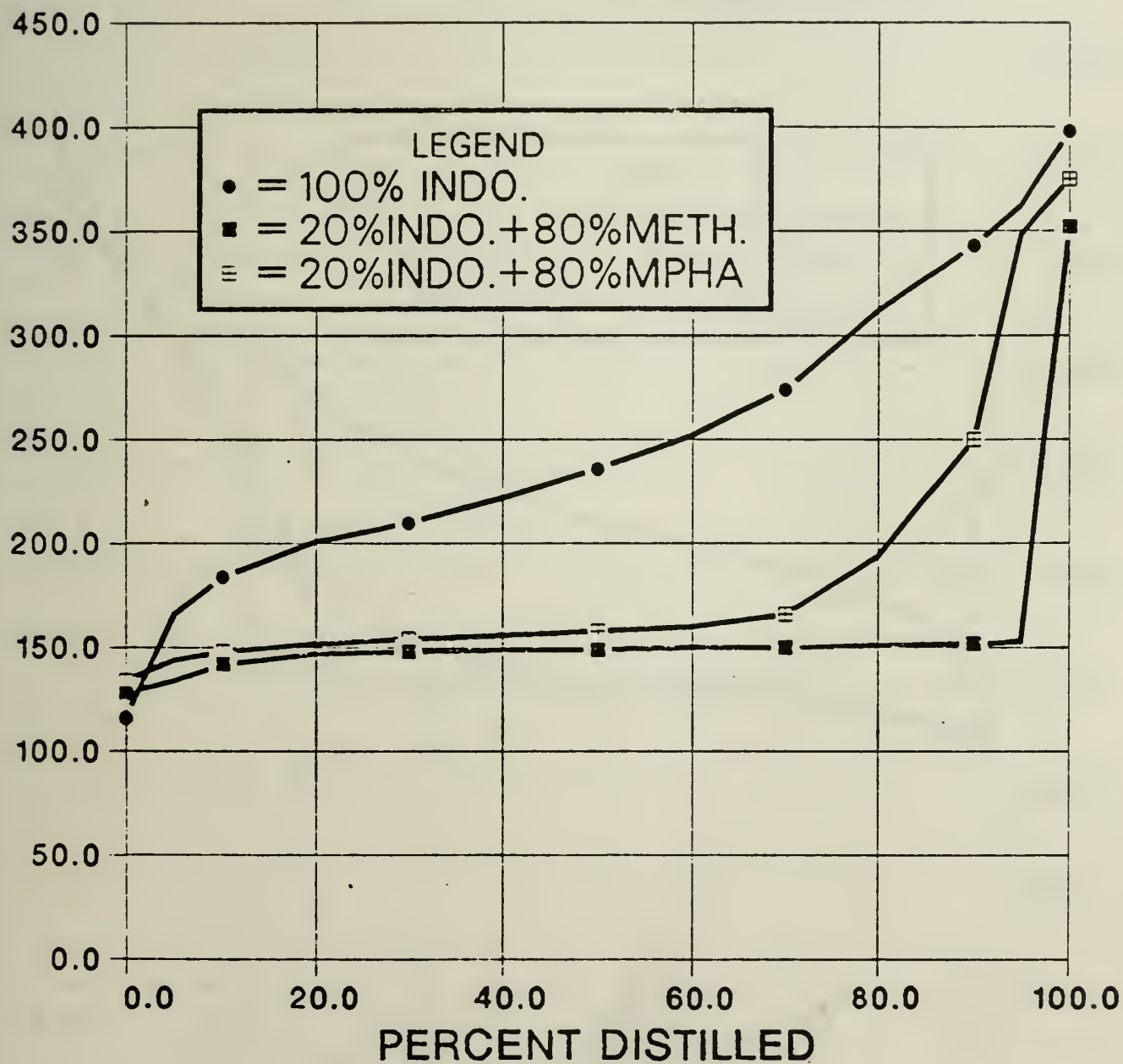


Fig. 5.2 Distillation Properties of 20% Blends.

DISTILLATION CHARACTERISTICS
INDO.-METH. & INDO.-MPHA BLENDS

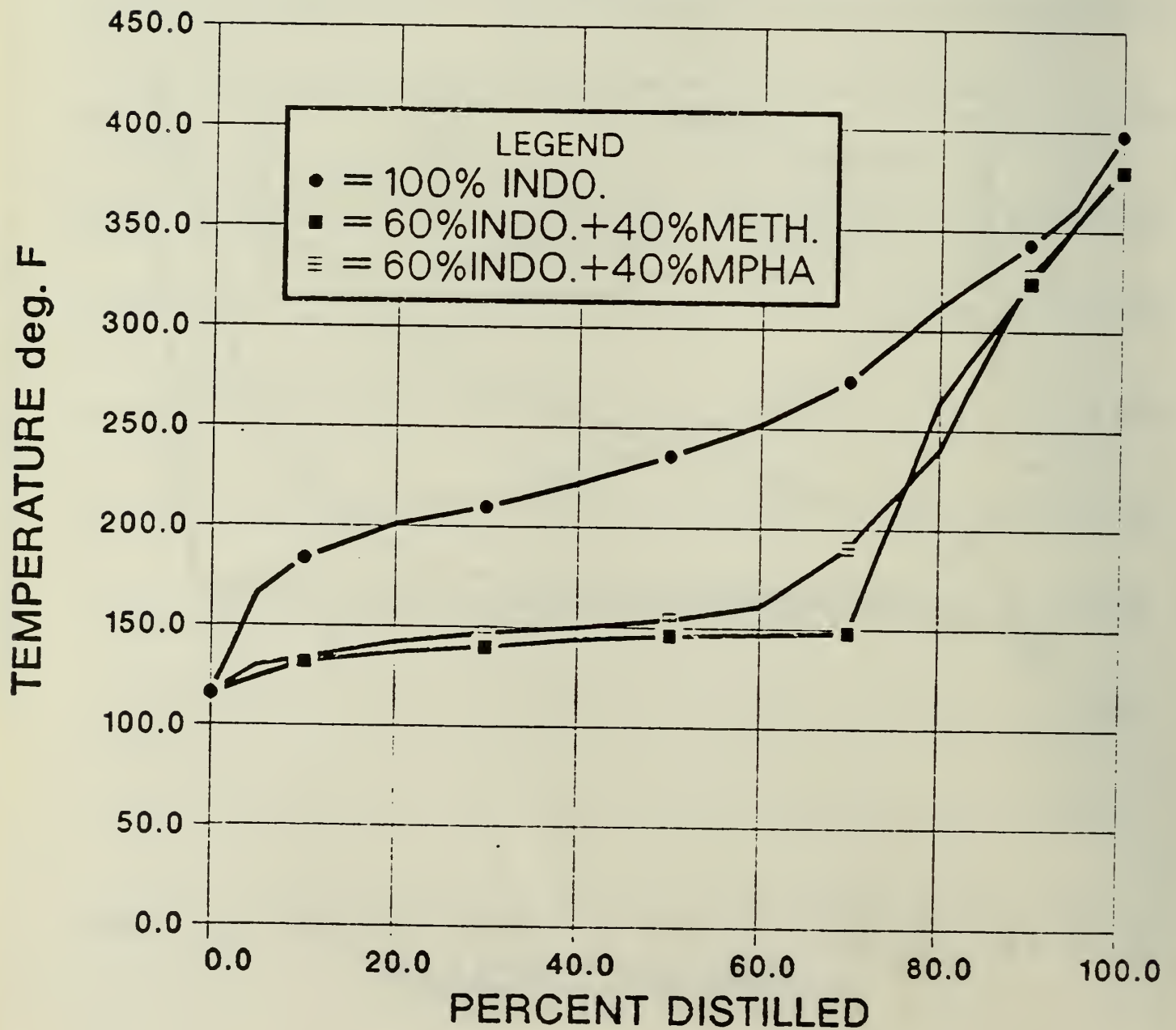


Fig. 5.3 Distillation Properties of 40% Blends.

DISTILLATION CHARACTERISTICS
INDO.-METH. & INDO.-MPHA BLENDS

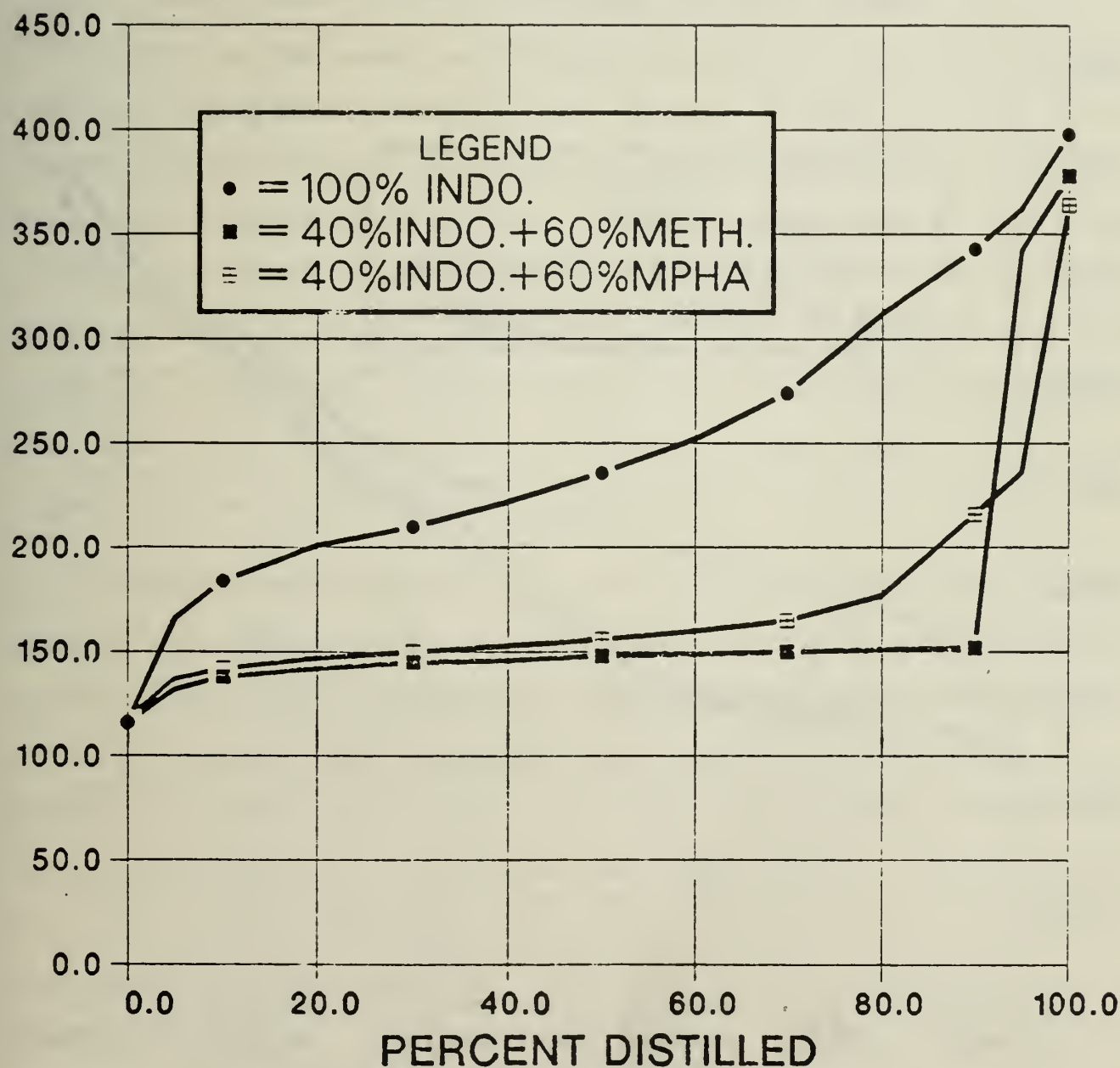


Fig. 5.4 Distillation Properties of 60% Blends.

DISTILLATION CHARACTERISTICS INDO.-METH. & INDO.-MPHA BLENDS

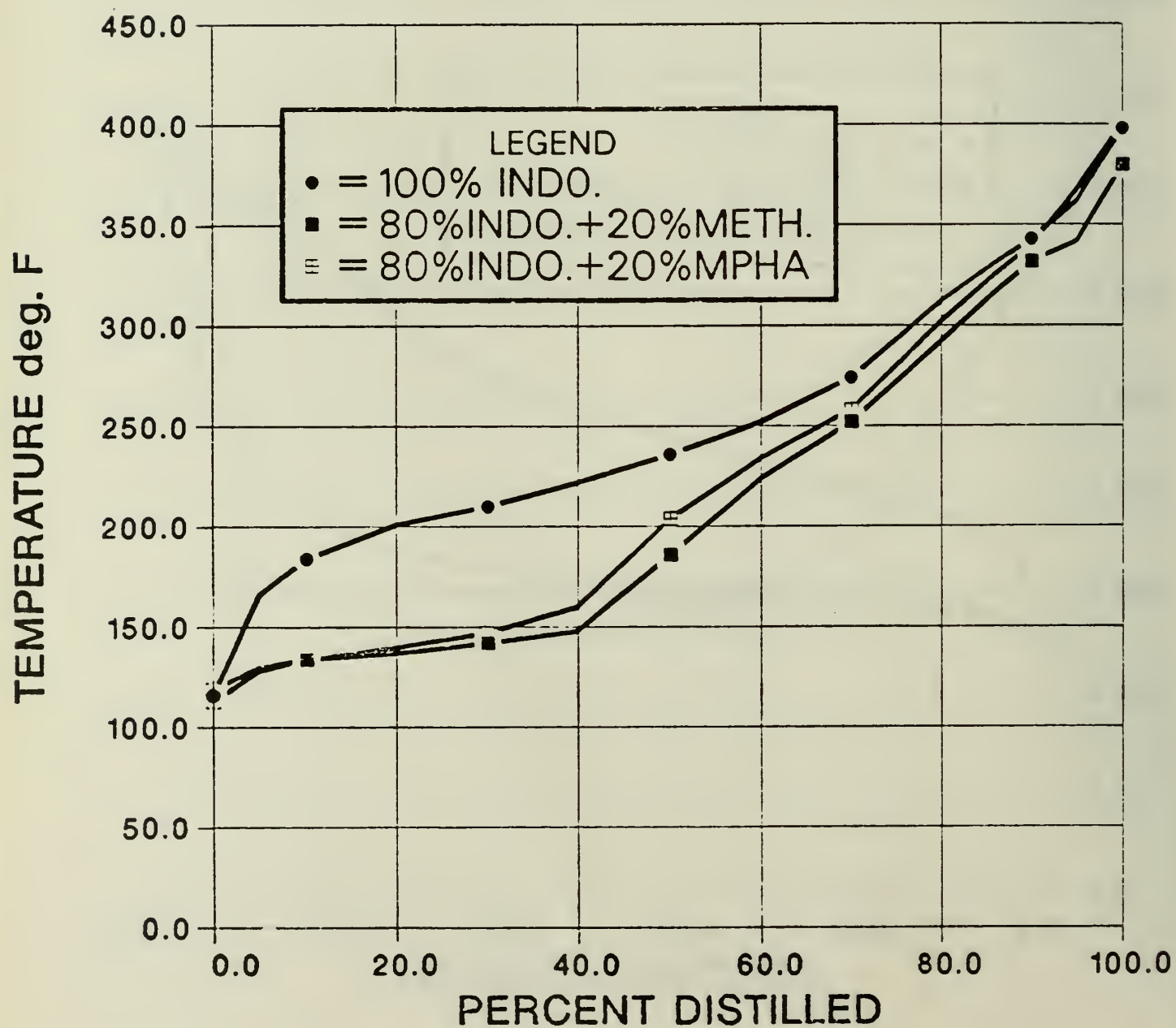


Fig. 5.5 Distillation Properties of 80% Blends.

along the curve are necessary to characterize a fuel, namely, the initial boiling point, 10 percent maximum, 50 percent minimum, 50 percent maximum, 90 percent maximum, and the end point. Depending upon the volatility class, the maximum 10 percent limit varies from 122F to 158F, minimum 50 percent is 170F, maximum 50 percent varies from 230F to 250F, maximum 90 percent varies from 365F to 374F and end point is 437F. Since the 50 percent minimum limit is 20 to 25 degrees above the constant boiling point temperature, that part of the distillation specification will be difficult to meet using indolene-methanol blends. As shown in Fig.5.2, the 20 percent alcohol blend is within the specified limit. The 40 percent blend, Fig.5.3, exceeds the limit. However, by modifying the formulation of MPHA, it may be possible to meet this part of the specification upto 70 percent blend level.

The shape and level of the distillation curve affects gasoline engine performance. The characteristic indicating the cold starting quality of a fuel is represented by 0 to 20 percent evaporated. A highly volatile 0 to 20 percent fuel will aid cold starting, but may tend to cause vapor lock at high temperatures. The warm-up characteristics are related to the temperature at which 50 percent of the fuel is vaporized. A low 50 percent point yields better performance, but with an increase in the tendency to cause carburetor icing. Fuel economy relates to heating value, and density, which may also correlate somewhat with the 90 percent point. Low back-end volatility may lead to crankcase dilution, deposits, gum, and spark plug fouling.

Energy Density

Fuel economy is usually expressed in terms of BSFC for engines. The BSFC is a function of the energy-mass density, which affects the conversion efficiency of the chemical energy to useful mechanical work. The energy density, or heating value of a fuel is a measure of the amount of energy released as a result of the complete combustion of the fuel. The higher heating value (water formed by combustion is in vapor form) is measured using an adiabatic bomb calorimeter. The heating values of indolene, indolene-methanol, and indolene-MPHA blends were measured. Fig.5.6 shows the heating values of the indolene-alcohol blends. The measured heating value decreases as the concentration of alcohol increases in the blend. The heating value of indolene-MPHA blend is higher than indolene-methanol blend at all blend levels.

The energy-volume density, BTU/gal, is directly proportional to the product of the energy-mass density and the specific gravity. Since the specific gravity of MPHA is similar to that of methanol, the energy-volume density of indolene-MPHA blends also be greater than that of indolene-methanol blends.

The heating values of blends are generally predicted from the pure component properties by equation 5.1. Figure 5.7 shows the blend heating values predicted from its pure components using this equation, and the measured values, for indolene-MPHA blends.

$$HVM = MI * HVI + MA * HVA \quad (5.1)$$

where HVM = Heating value of mixture

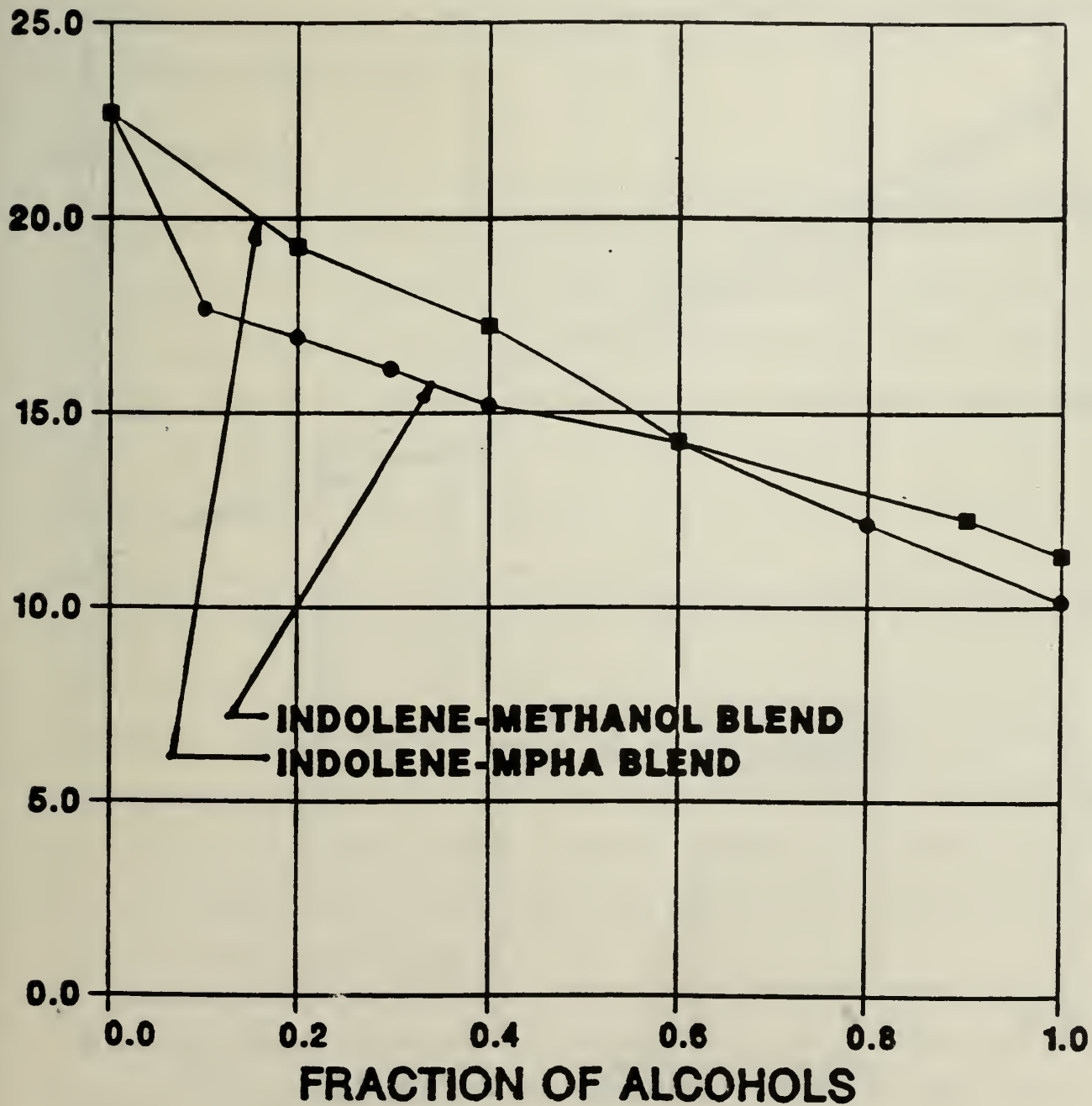
MEASURED HIGHER HEATING VALUE**INDO.-METH. & INDO.MPHA BLEND**

Fig. 5.6 Measured Higher Heating Values of Indolene-Alcohols Blends

MEASURED & CALCULATED HEATING VALUE INDOLENE-MPHA BLENDS

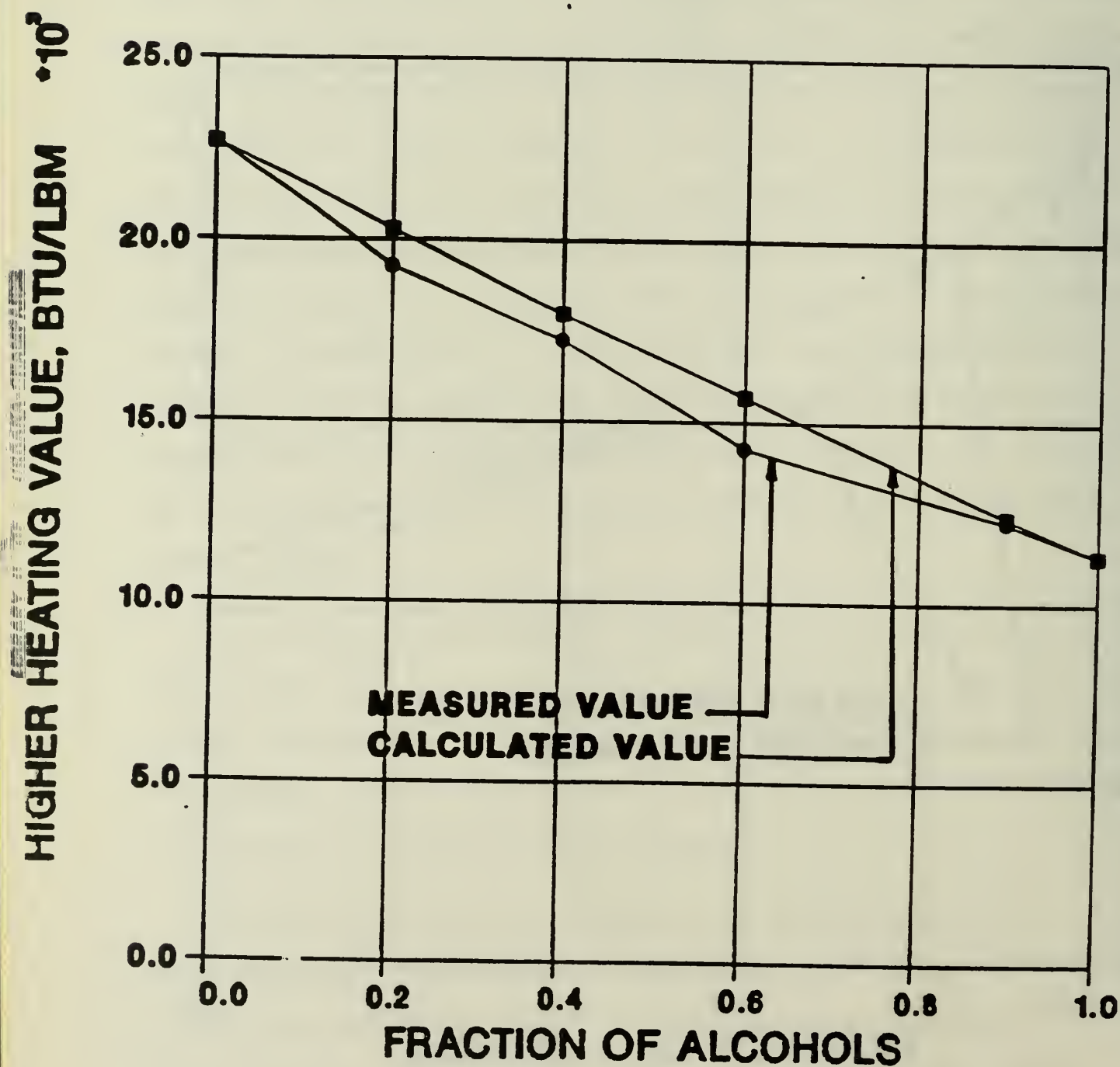


Fig. 5.7 Measured & Calculated Higher Heating Value of Indolene-MPHA Blends.

MI = Mass fraction of indolene

HVI = Heating value of indolene

MA = Mass fraction of alcohol

HVA = Heating value of alcohol

There is a good agreement between measured and predicted values for indolene-MPHA blends.

Water Tolerance

Fig.5.8 shows the water tolerance of the indolene-alcohol blends. The solubility of alcohols in indolene when water is present even in minute quantities is limited at room temperature. When small amounts of water added to blends of alcohol with indolene, hydrogen bonds form between the water and alcohol molecules, and the blends separate into two phases. Paraffinic hydrocarbons predominate in the upper phase, while the lower phase consists primarily of alcohols and water. Water tolerance is also significantly decreased at lower temperatures. At room temperature, as the alcohol concentration increases, water tolerance increases. The 90 percent methanol blend has a water tolerance of 8.25 percent by volume, compared to 0.25 percent at 10 percent blend. The 90 percent MPHA blend has about 50 percent higher water tolerance compared to the 90 percent methanol blend at room temperature.

Solubility of methanol and indolene in the presence of water can be one of the most difficult problems attending the use of these blends as motor fuels. The fuel distribution system, refinery, pipeline, and terminal tanks now used are not necessarily designed

WATER TOLERANCE

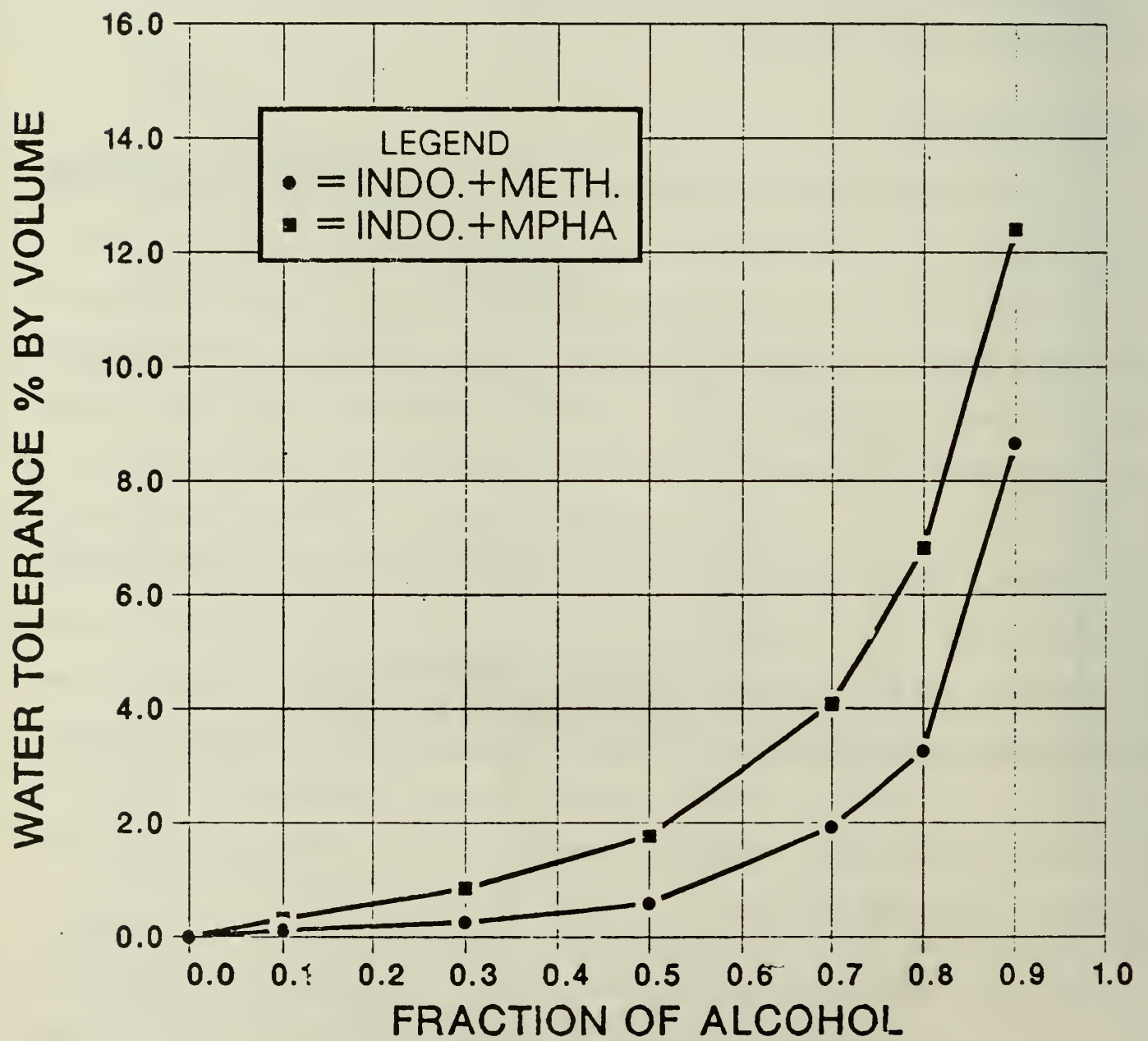


Fig. 5.8 Water Tolerance

to fully exclude rainwater from coming into contact with the fuel. Condensation in the storage tanks can cause additional water contamination. Using MPHA-gasoline blends instead of methanol-gasoline blends increases the water tolerance, and may be used without making major modifications to the existing fuel distribution and delivery system.

Flash Point

The flash point is the lowest temperature at which the vapors arising from a fuel will momentarily ignite as a flame is swept across its surface. It is used to assess the flammability of a fuel and to establish shipping and safety regulations. The flash points of methanol and MPHA fuels were measured using a Pensky-Martens closed up tester, following the ASTM D93 method. The flash point of pure methanol was 50F, and that of MPHA, 49F.

API Gravity and Specific Gravity

The API gravity is defined by an arbitrary scale whose units are degrees API, and is measured at 15.5 deg.C. The API gravity of indolene-methanol and indolene-MPHA blends were measured using Fisher brand Petroleum hydrometers. Specifications for API gravity hydrometers have been established by ASTM. Fig.5.9 shows the API gravity of indolene-methanol and indolene-MPHA blends. In general, as the alcohol content increases, API gravity decreases. Indolene-methanol has a slightly higher API gravity than indolene-MPHA.

The specific gravity is found by dividing the density of the fuel

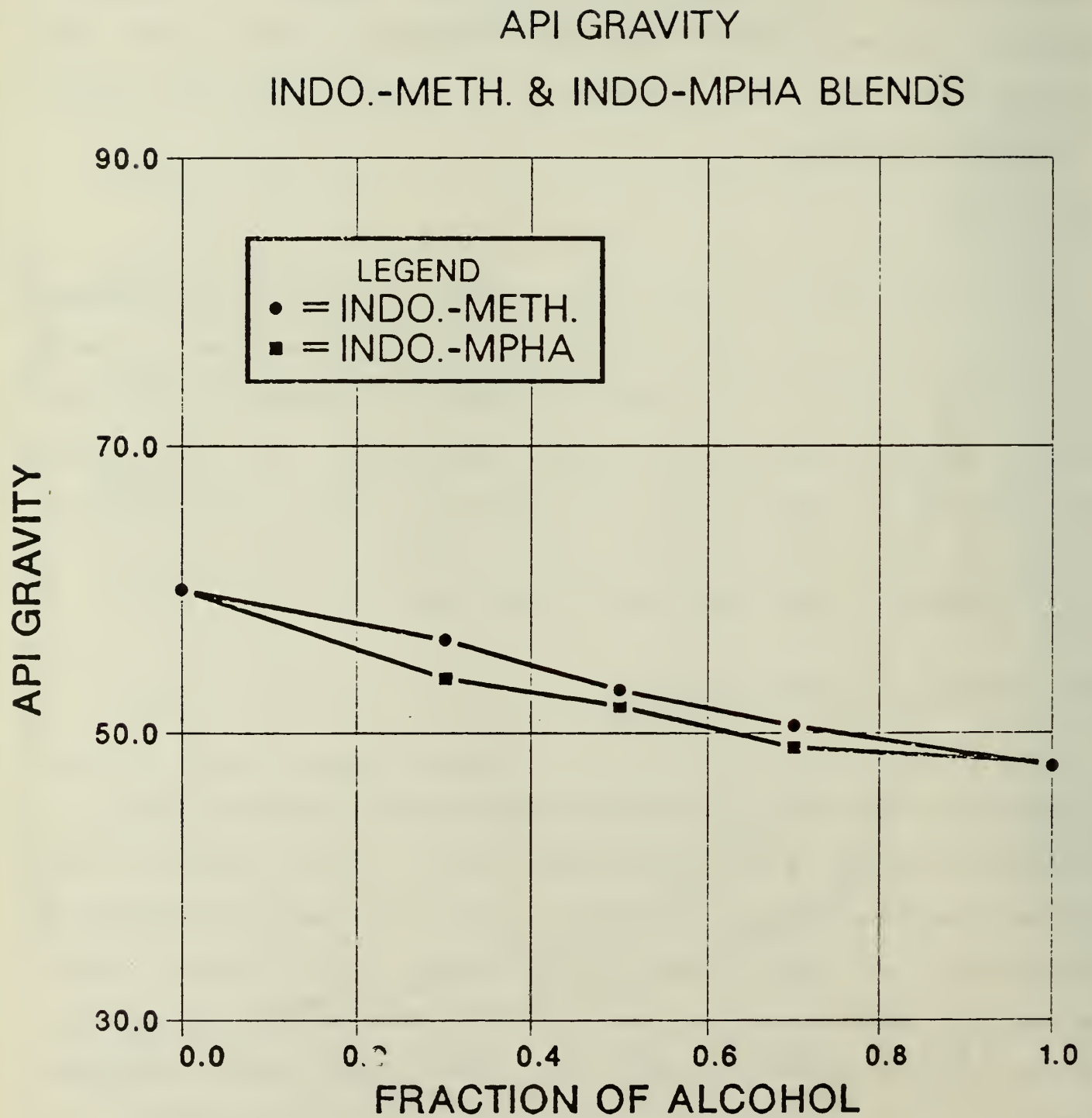


Fig. 5.9 API Gravity

by the density of water. The specific gravity is related to API gravity as shown in equation 5.2.

$$\text{Degree API} = (141.5/\text{specific gravity}) - 131.5 \quad (5.2)$$

Fig.5.10 shows the specific gravity of indolene-methanol and indolene-MPHA blends. As the alcohol concentration increases, the specific gravity increases slightly. In general, indolene-MPHA has a higher specific gravity than indolene-methanol blend.

Stoichiometry

Stoichiometry and combustion are related to the H/C ratio of the fuel. In general, for indolene, H/C is 1.86, 4.0 for methanol, and 3.269 for MPHA. The stoichiometric air-fuel ratio depends upon the H/C ratio of the fuel. The stoichiometric air-fuel ratio is 14.56 for pure indolene, 6.43 for methanol, and 7.4468 for MPHA. Fig.5.11 shows the variation of stoichiometric air-fuel ratio as a function of alcohol concentration. As the alcohol concentration increases, stoichiometric air-fuel ratio decreases. The stoichiometric air-fuel ratio of indolene-MPHA blends is higher than those of indolene-methanol blends.

PERFORMANCE PARAMETERS

Effect of Alcohol on MBT Spark Advance

Figures 5.13-5.16 show the effect of varying the methanol and MPHA concentrations on the MBT spark advance, at different compression ratios. At any fixed compression ratio, as the alcohol concentration increases, the MBT spark advance decreases, or shifts

SPECIFIC GRAVITY
INDO.-METH. & INDO.-MPHA BLENDS

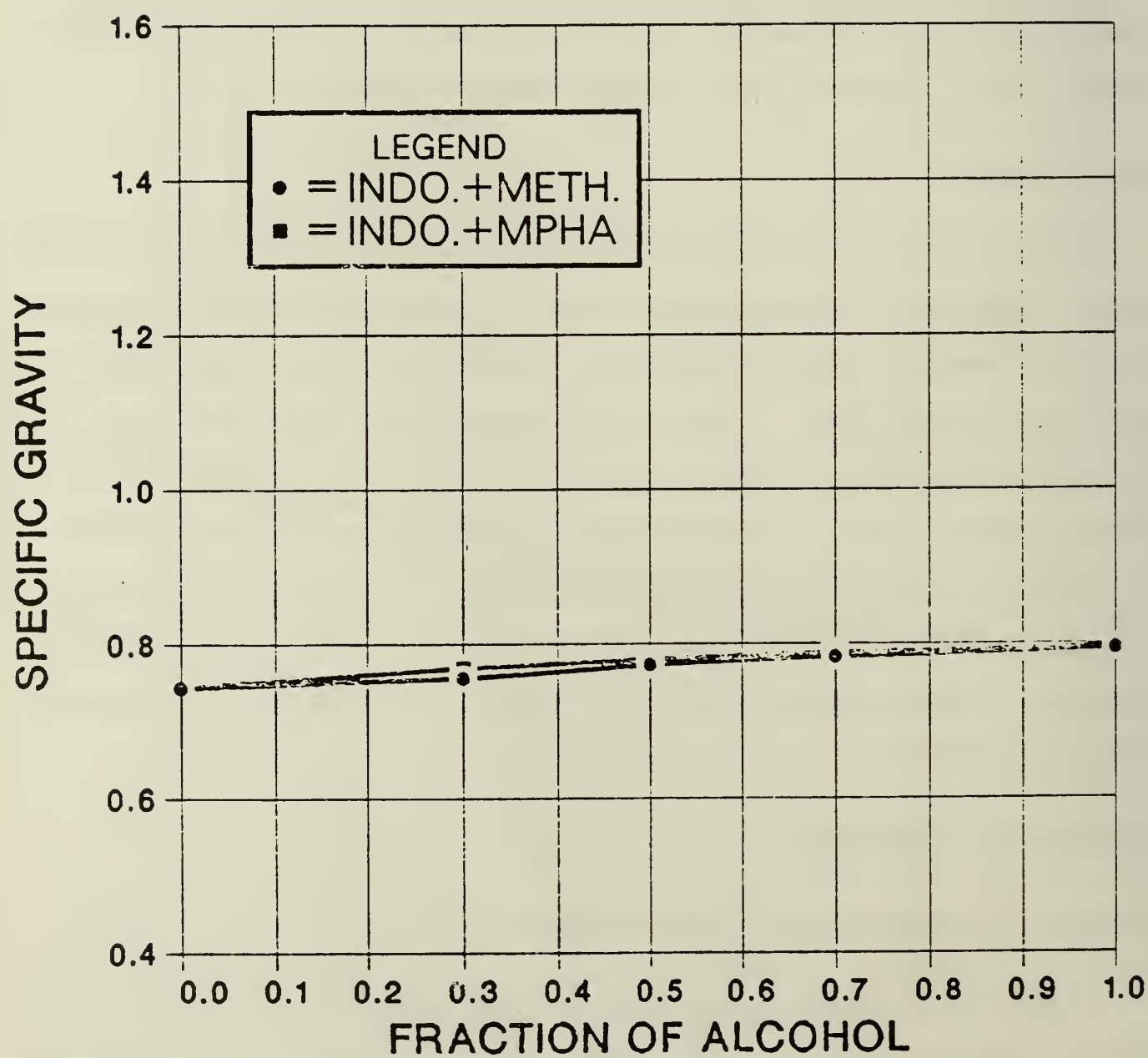


Fig. 5.10 Specific Gravity

STOICHIOMETRIC A/F RATIO
INDO.-METH. & INDO.-MPHA BLENDS

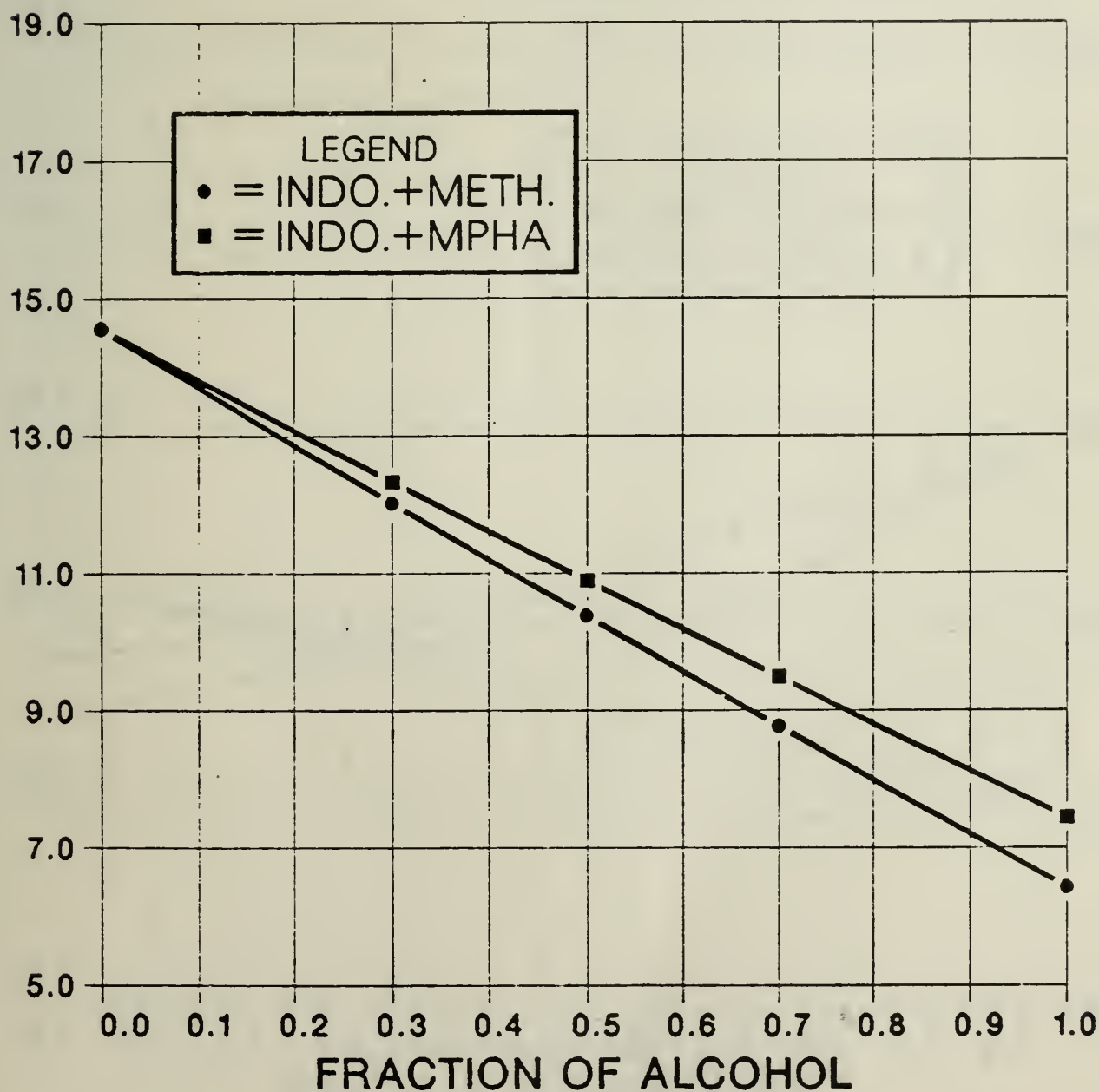


Fig. 5.11 Stoichiometric A/F Ratio

MBT SPARK ADVANCE AT CR=5.0
INDO.-METH. & INDO.-MPHA BLENDS

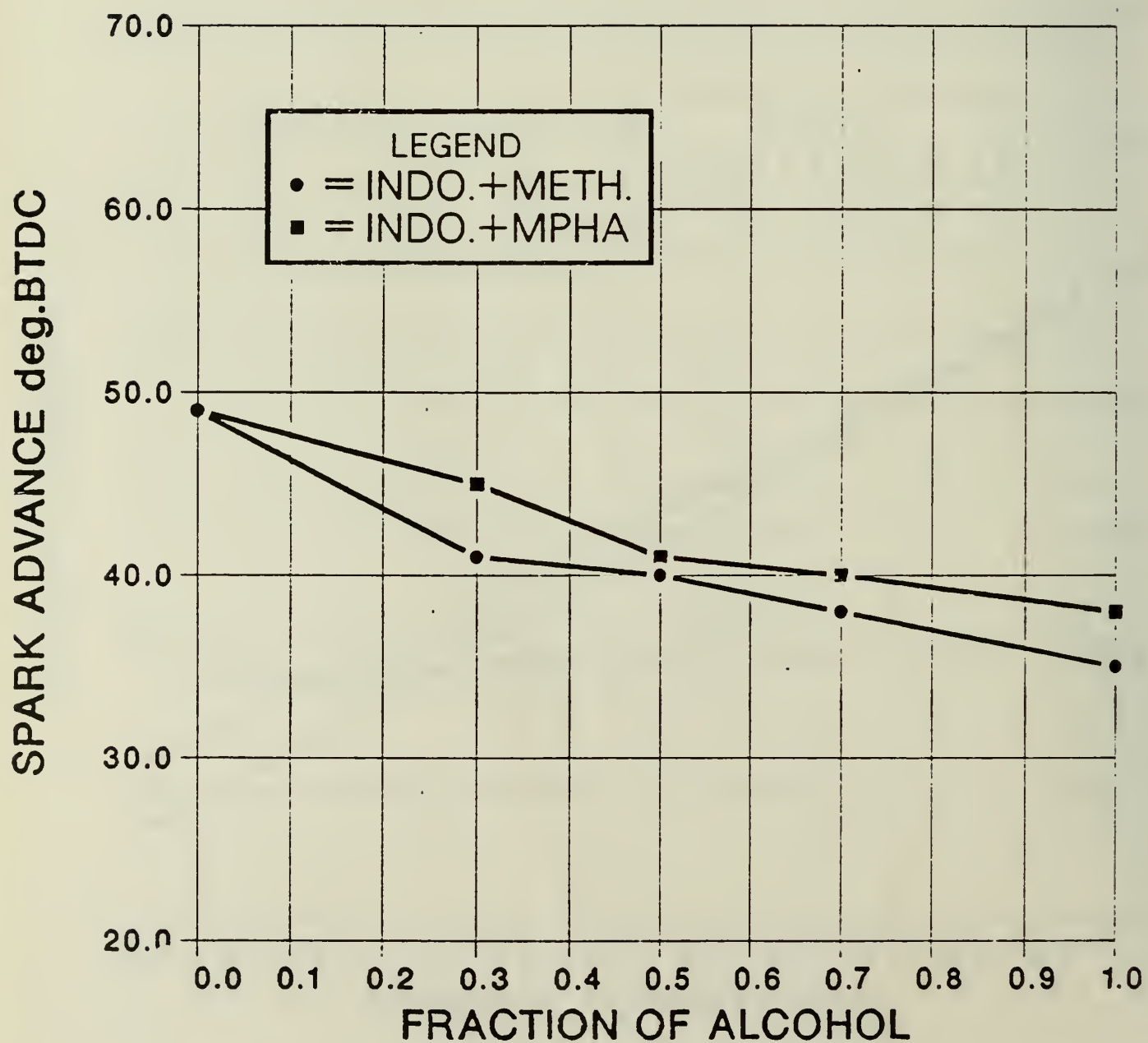


Fig. 5.13 MBT Spark Advance at CR=5:1

MBT SPARK ADVANCE AT CR=6.0
INDO.-METH. & INDO.-MPHA BLENDS

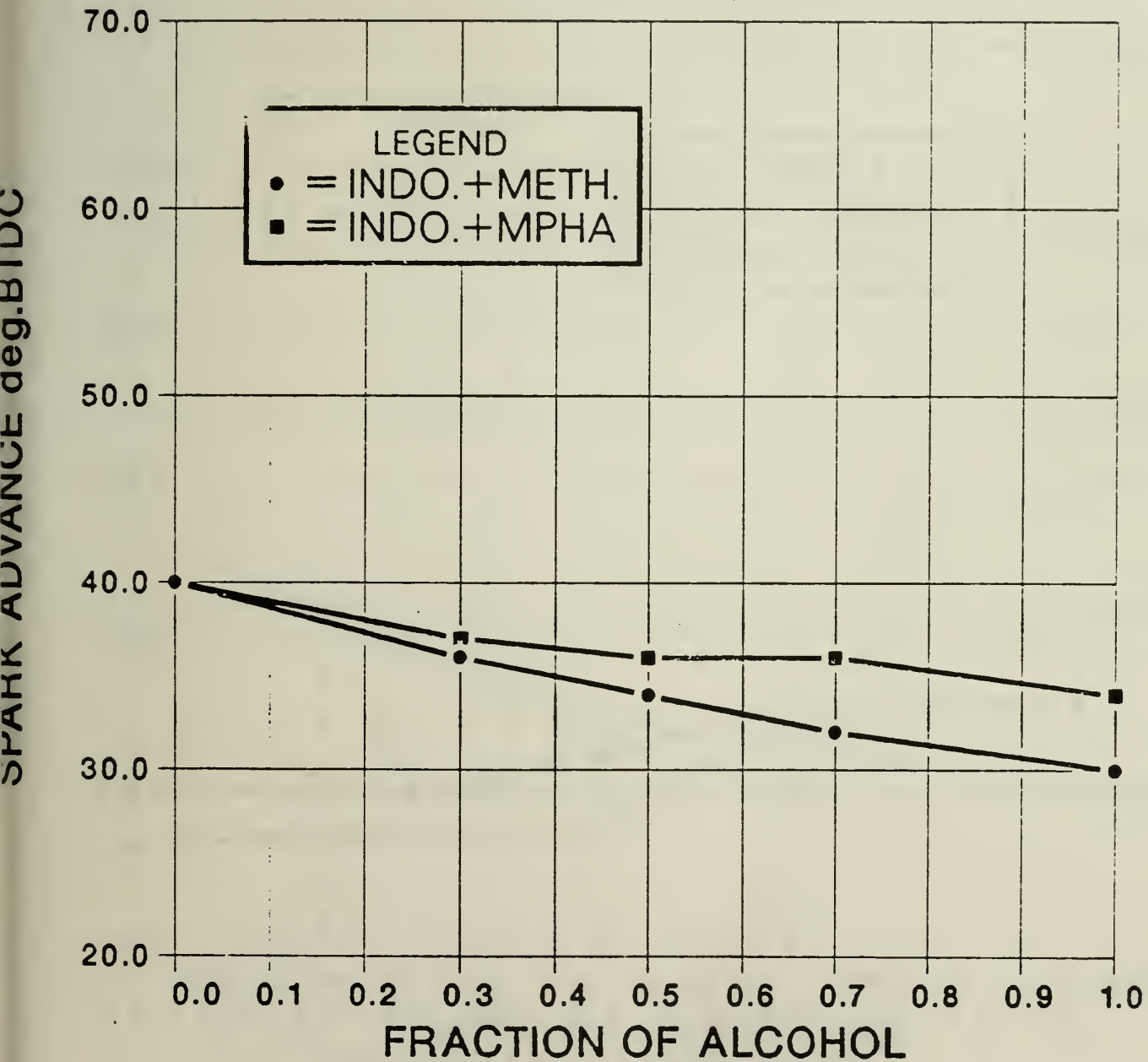


Fig. 5.14 MBT Spark Advance at CR=6:1

MBT SPARK ADVANCE AT CR=7.0
INDO.-METH. & INDO.-MPHA BLENDS

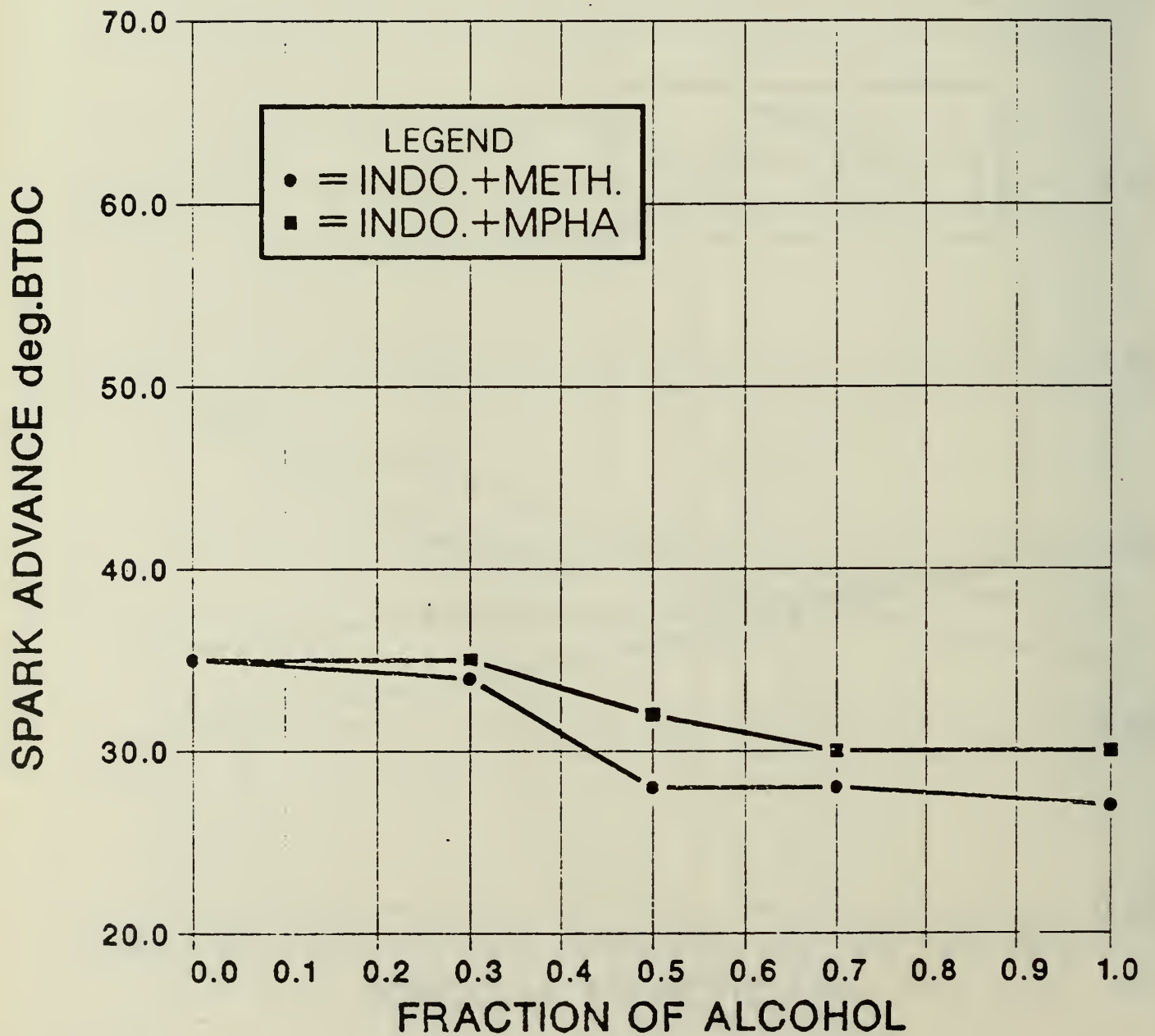


Fig. 5.15 MBT Spark Advance at CR=7:1

MBT SPARK ADVANCE AT KLCR
INDO.-METH. & INDO.-MPHA BLENDS

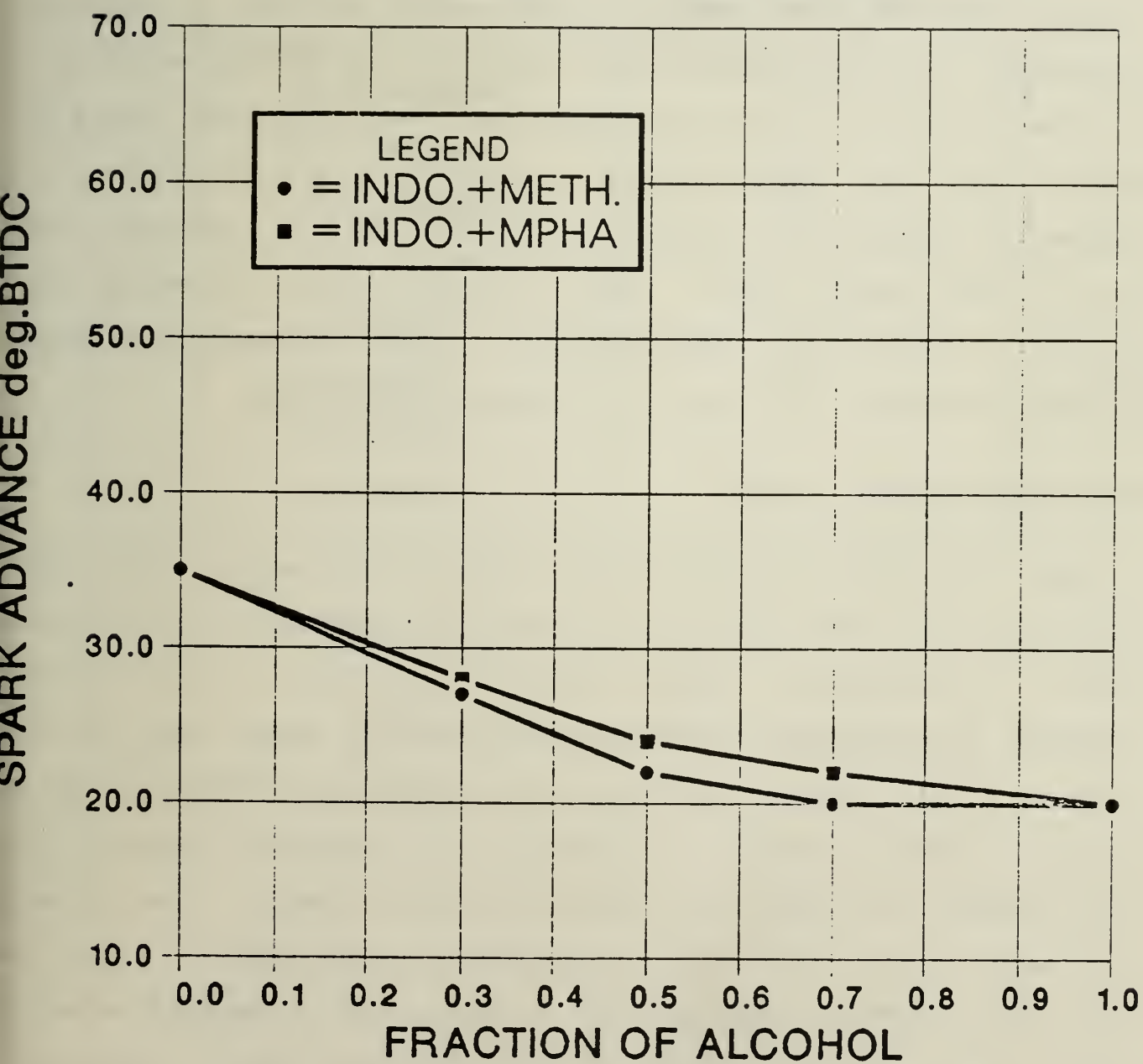


Fig. 5.16 MBT Spark Advance at CR=KLCR

towards the top dead center. This is an indication of a higher burning velocity of indolene-alcohol blends compared to pure indolene. At any compression ratio, the MBT spark advance of indolene-MPHA blends is higher than that of indolene-methanol blends. The MBT spark advance is controlled by the thermodynamic properties, such as temperature, pressure and burning velocity at one specific engine operating condition. With the same intake air temperature, the temperature of the air-fuel mixture during the compression process is related to the latent heat of the fuel. MPHA has a lower latent heat, and a higher burning velocity than methanol. However, the combined effect on spark advance is a higher MBT spark advance in the case of indolene-MPHA blends.

Effect of Alcohol on KLCR

Table 5.2 shows the KLCR as a function of methanol and MPHA concentration in the blend. As the methanol concentration increases from 0 to 40 percent, KLCR increases from 7:1 to 10:1. Further increase in methanol concentration does not cause any further increase in the KLCR. The variation of KLCR with MPHA fraction is similar to that of methanol. However, the knocking intensity with indolene-MPHA was lower than indolene-methanol blends. The maximum compression ratio that can be obtained in this type of engine is 10:1. The KLCR of indolene-MPHA is higher than indolene-methanol blend. The actual magnitude of the KLCR could not be obtained, however, due to the limitation of the engine.

TABLE 5.2

KNOCK LIMITED COMPRESSION RATIO

Volume Fraction of Alcohol in Indolene	Knock Limited Compression Ratio
0.3 methanol	8.5
0.5 methanol	10
0.7 methanol	10
1.0 methanol	10
0.3 MPHA	8.87
0.5 MPHA	10
0.7 MPHA	-
1.0 MPHA	-

Effect of Alcohol on Brake Power

Figures 5.17-5.20 show the brake power as a function of fraction of alcohols. In general, as the alcohol concentration increases, brake power increases. At any compression ratio, or any fraction of alcohol, the brake power of indolene-methanol is no different from that of indolene-MPHA.

Effect of Alcohol on Brake Specific Fuel Consumption

Figures 5.21-5.24 show the brake specific fuel consumption as a function of fraction of alcohols. In general, as the alcohol concentration increases, the brake specific fuel consumption increases. At any compression ratio, indolene-methanol has a higher BSFC compared to indolene-MPHA, specifically after 70 percent blend level. It is noted that BSFC is on a mass basis, (kg/kW-hr). With regards to BSFC, methanol is at a distinct disadvantage compared to MPHA because of its lower heating value.

Effect of Alcohol on Brake Thermal Efficiency

Figures 5.25-5.28 show the brake thermal efficiency as a function of alcohol fraction. At any compression ratio, the thermal efficiency of indolene-alcohol blend is higher than that of indolene. In general, the indolene-methanol blends have a higher thermal efficiency than indolene-MPHA blends. The brake thermal efficiency depends upon combustion period, cooling losses, chemical equilibrium losses, change in specific heat losses, and latent heat of vaporization. The thermodynamic analysis of the entire cycle is

BRAKE POWER AT CR=5.0
INDO.-METH. & INDO.-MPHA BLENDS

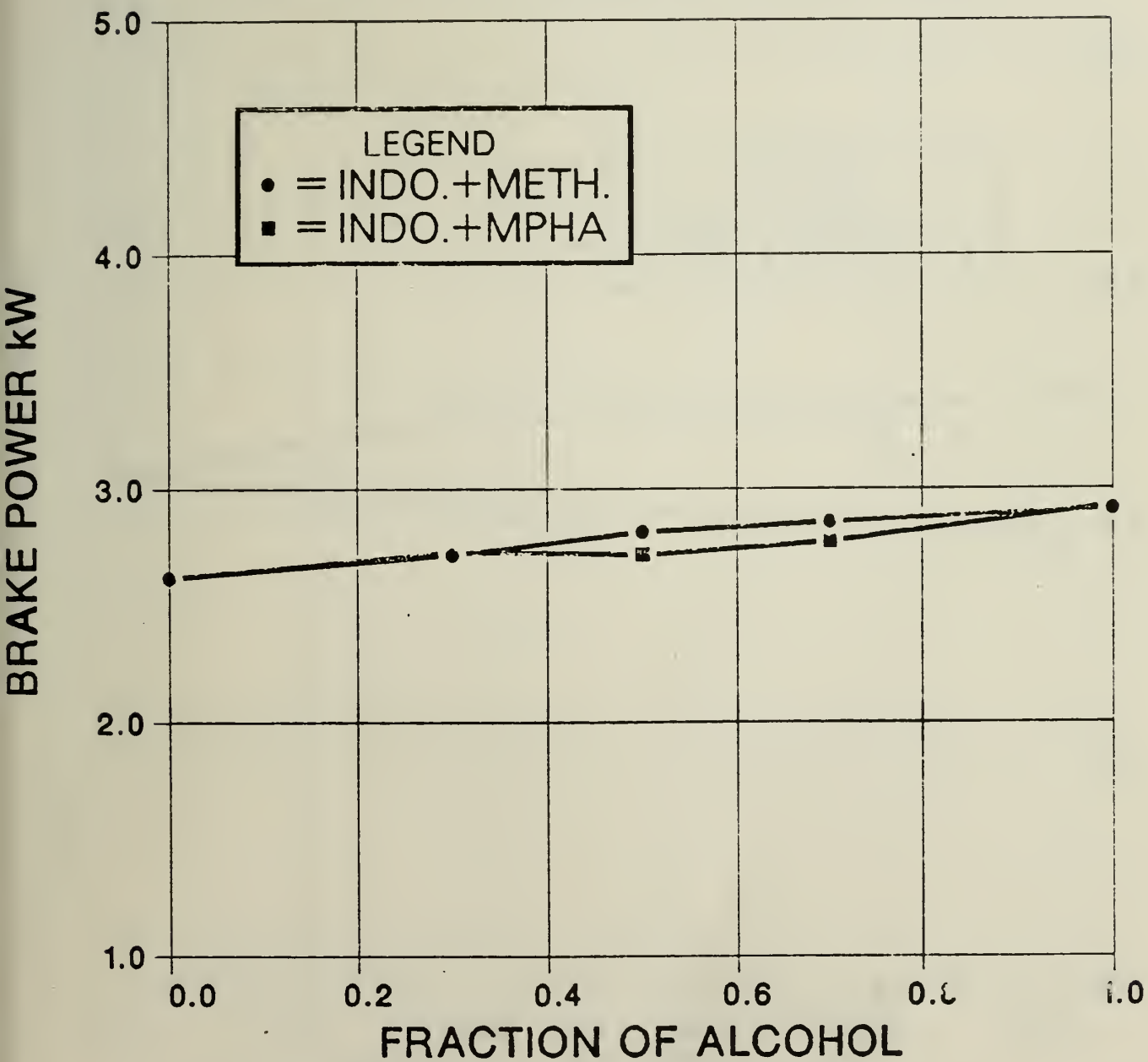


Fig. 5.17 Brake Power at CR=5:1

BRAKE POWER AT CR=6.0
INDO.-METH. & INDO.-MPHA BLENDS

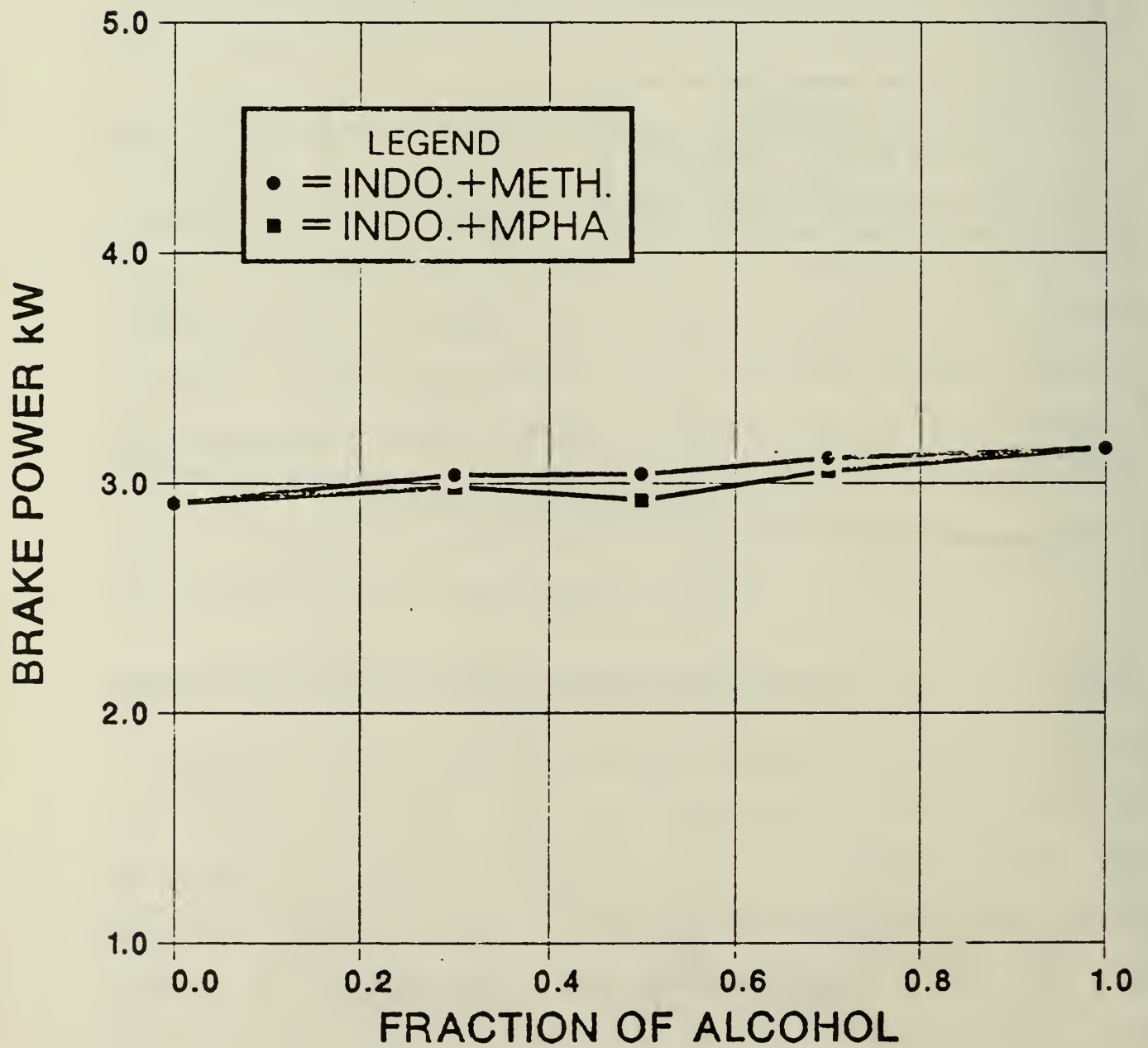


Fig. 5.18 Brake Power at CR=6:1

BRAKE POWER AT CR=7.0
INDO.-METH. & INDO.-MPHA BLENDS

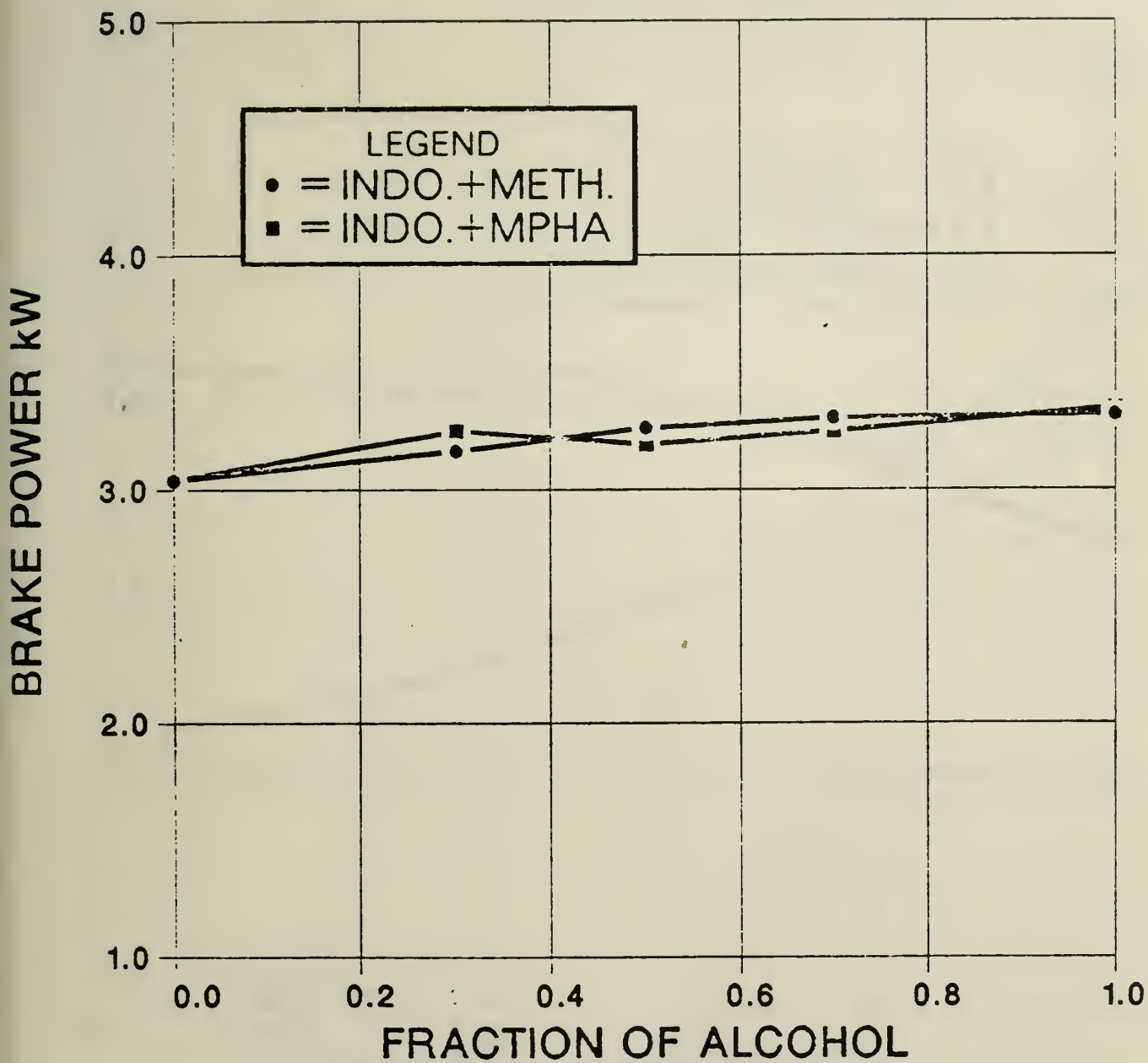


Fig. 5.19 Brake Power at CR=7:1

BRAKE POWER AT KLCR
INDO.-METH. & INDO.-MPHA BLENDS

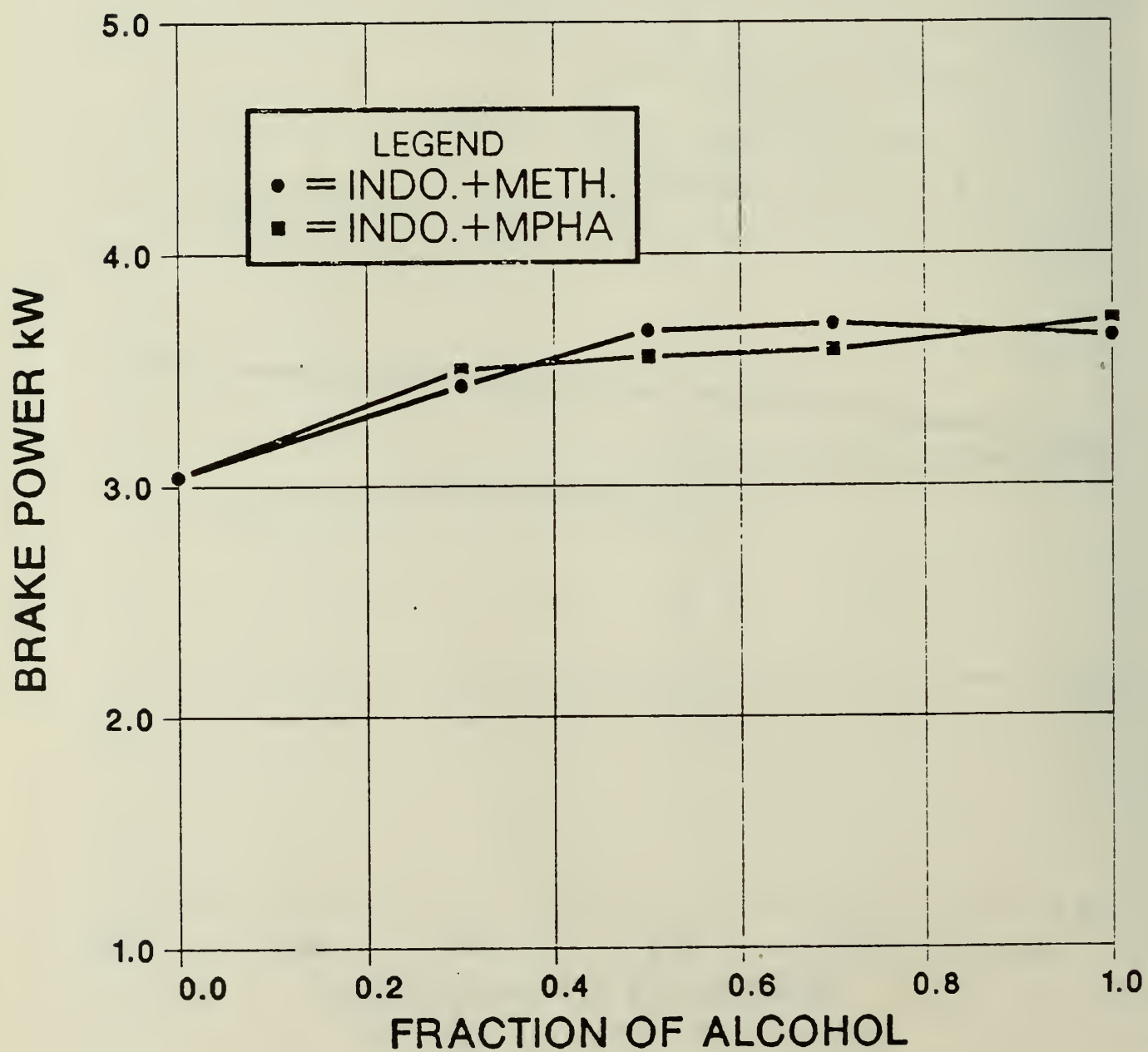


Fig. 5.20 Brake Power at CR=KLCR

FUEL CONSUMPTION AT C.R.=5.0
INDO.-METH. & INDO.-MPHA BLENDS

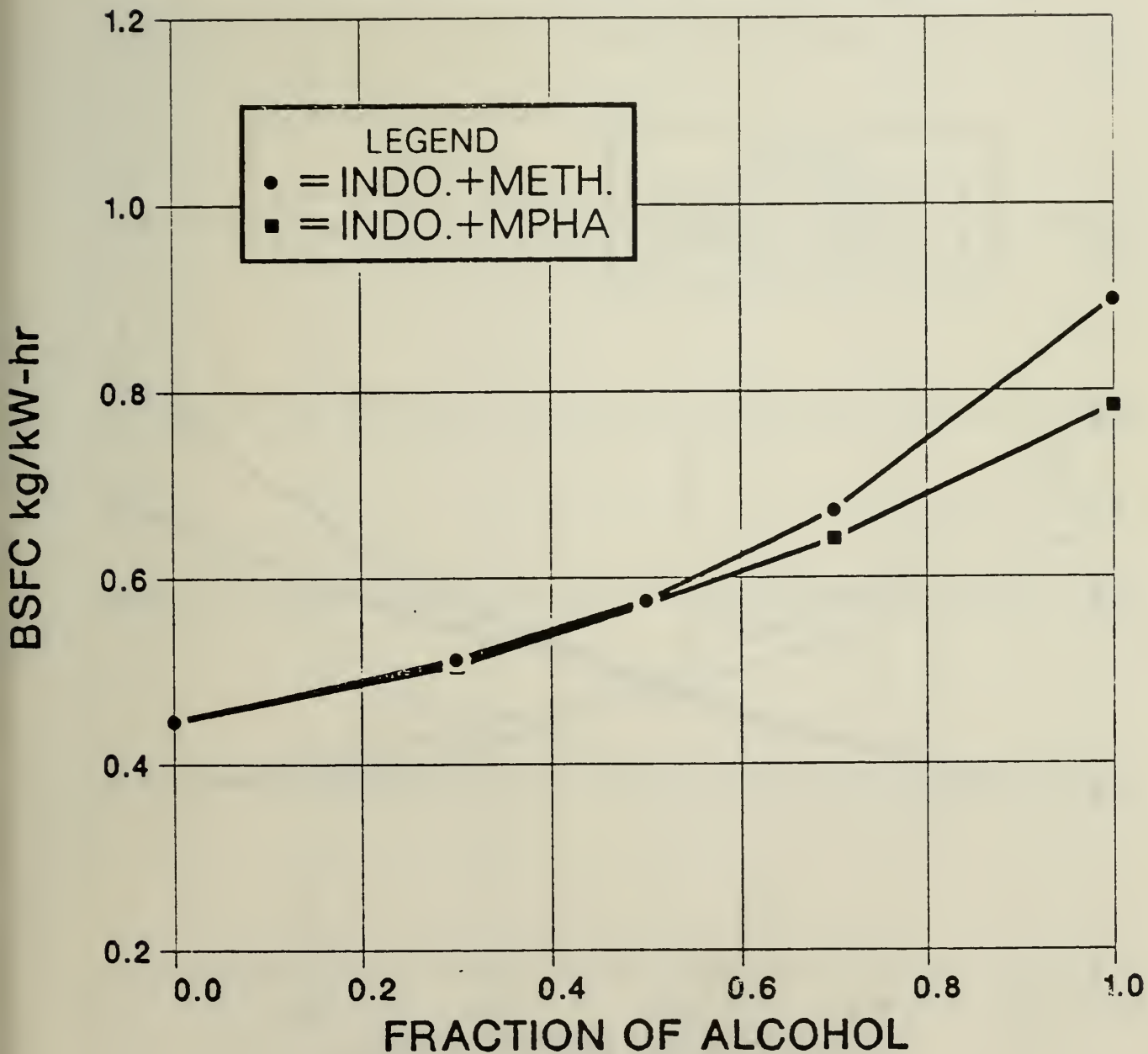


Fig. 5.21 BSFC at CR=5:1

FUEL CONSUMPTION AT C.R.=6.0
INDO.-METH. & INDO.-MPHA BLENDS

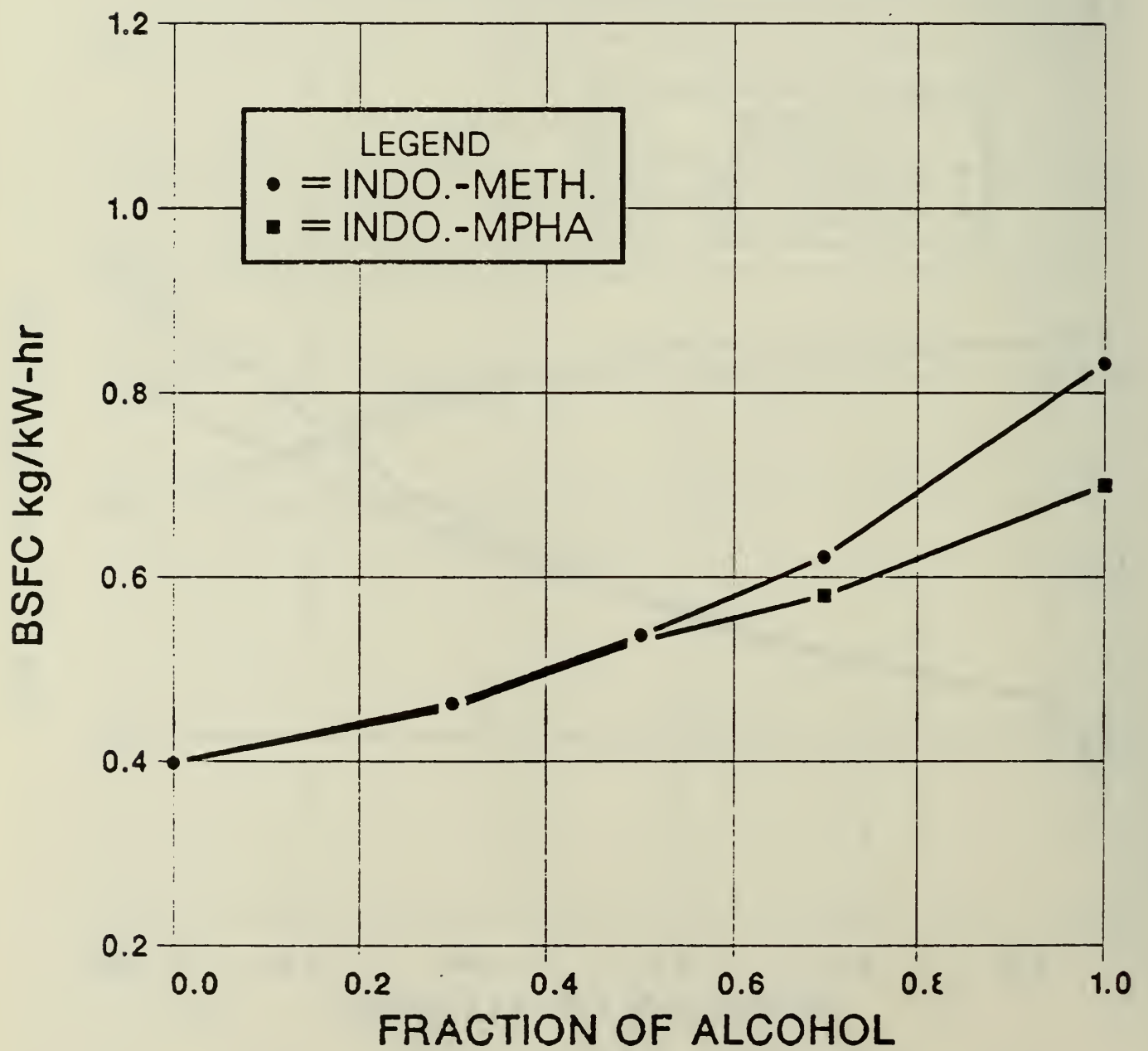


Fig. 5.22 BSFC at CR= 6:1

FUEL CONSUMPTION AT C.R.=7.0
INDO.-METH. & INDO.-MPHA BLENDS

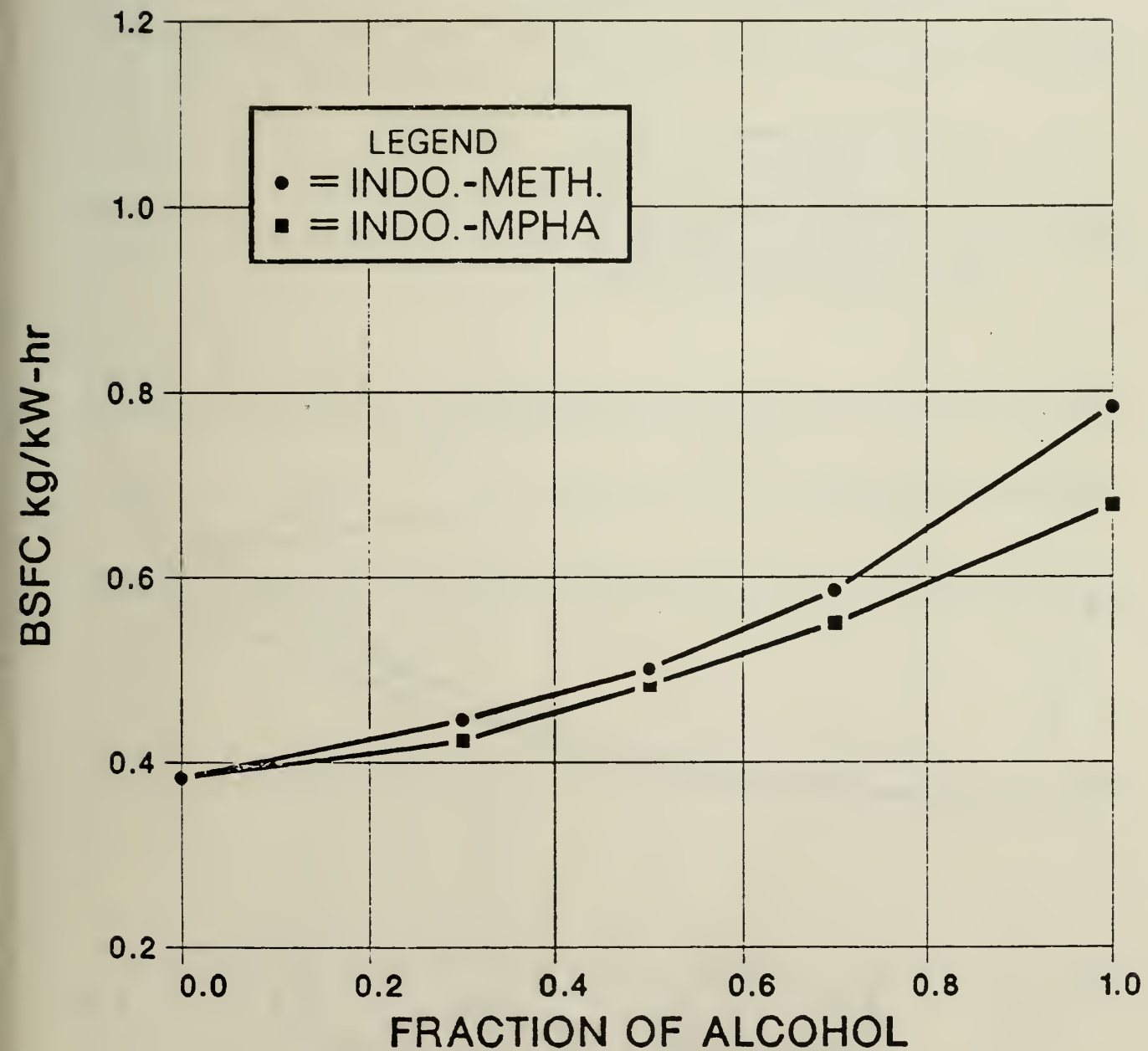


Fig. 5.23 BSFC at CR=7:1

FUEL CONSUMPTION AT KLCR
INDO.-METH. & INDO.-MPHA BLENDS

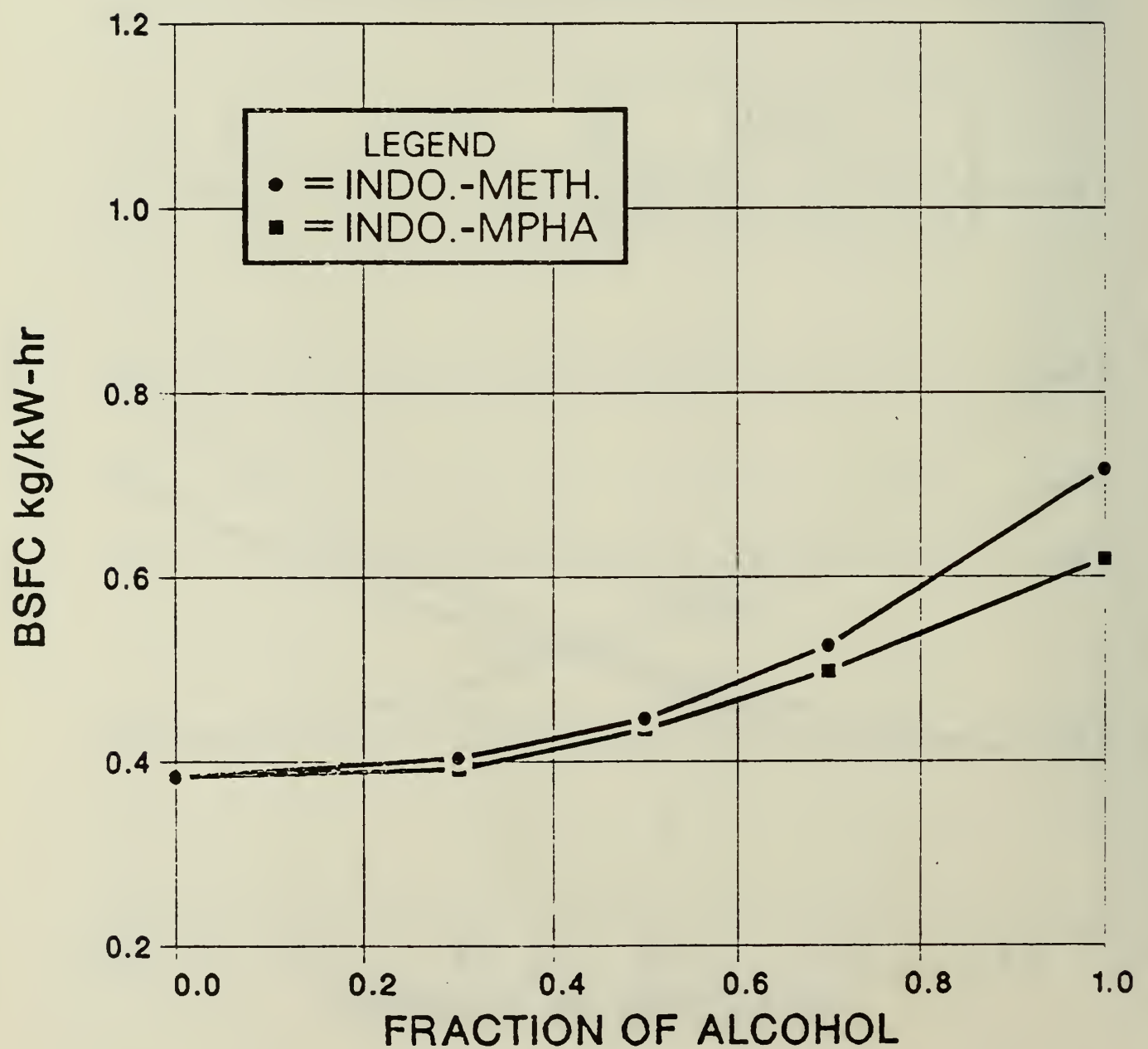


Fig. 5.24 BSFC at CR=KLCR

THERMAL EFFICIENCY AT C.R.=5.0
INDO.-METH. & INDO.-MPHA BLENDS

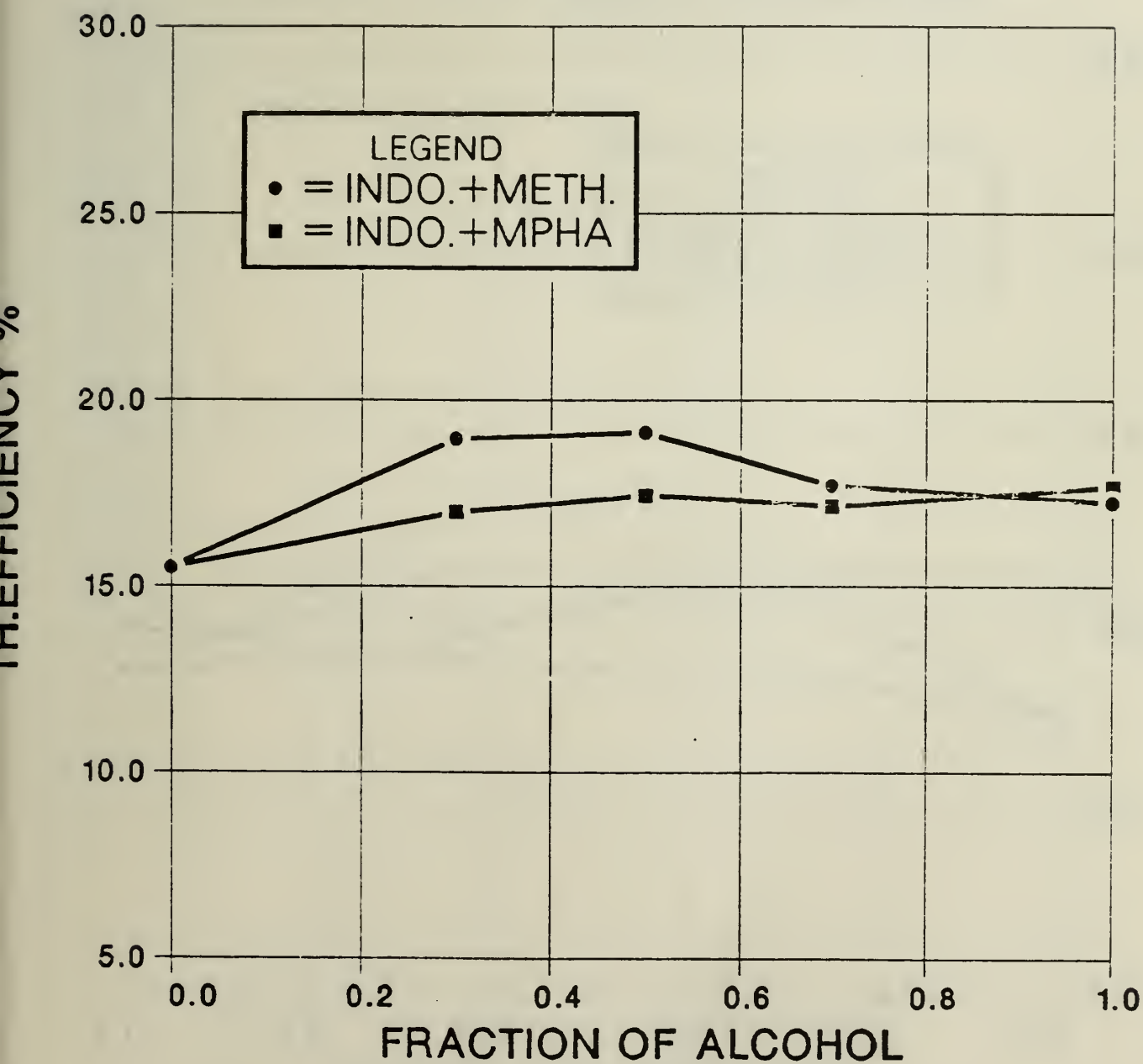


Fig. 5.24 Thermal Efficiency at CR=5:1

THERMAL EFFICIENCY AT C.R.=6.0
INDO.-METH. & INDO.-MPHA BLENDS

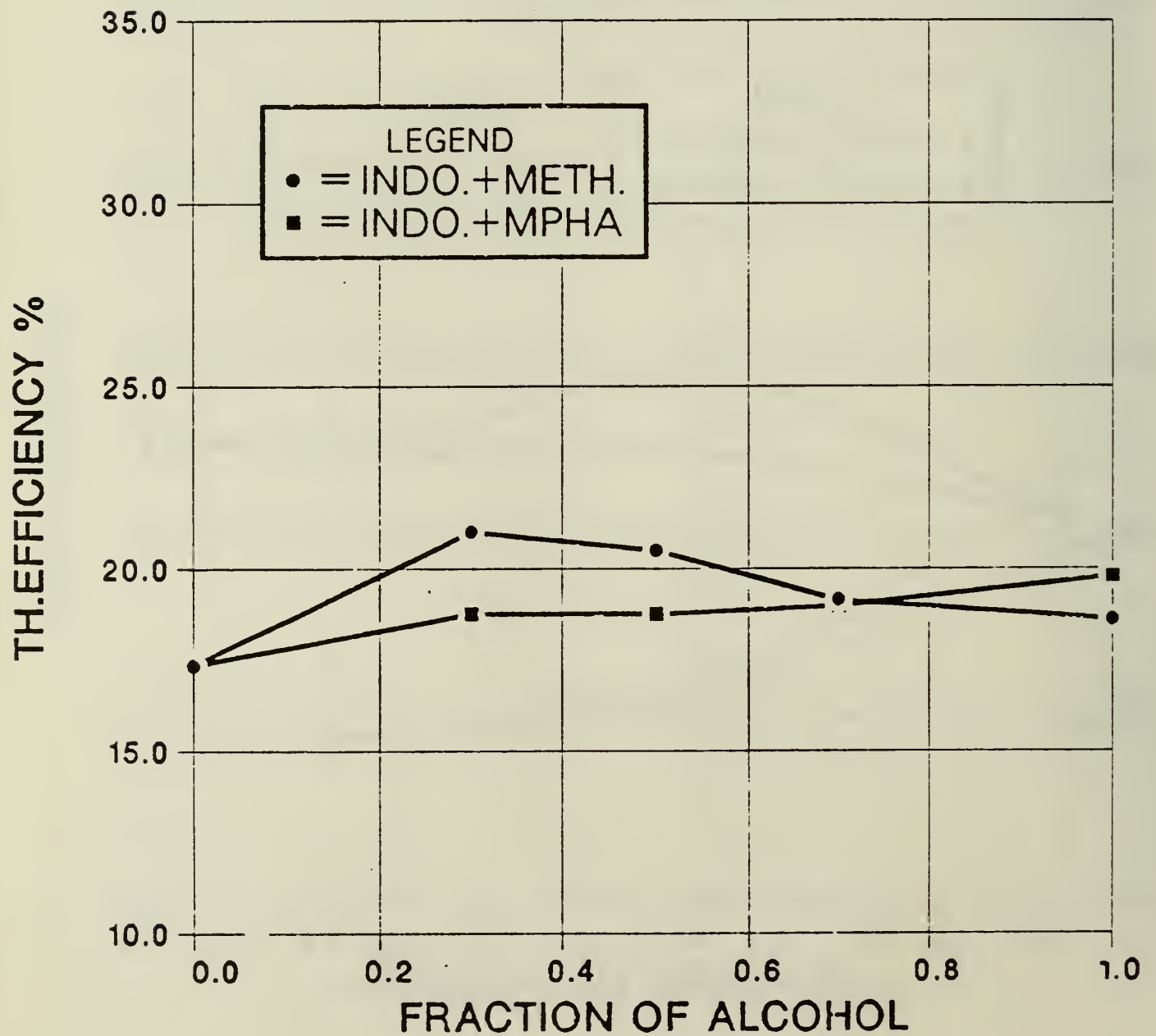


Fig. 5.26 Thermal Efficiency at CR=6:1

THERMAL EFFICIENCY AT C.R.=7.0
INDO.-METH. & INDO.-MPHA BLENDS

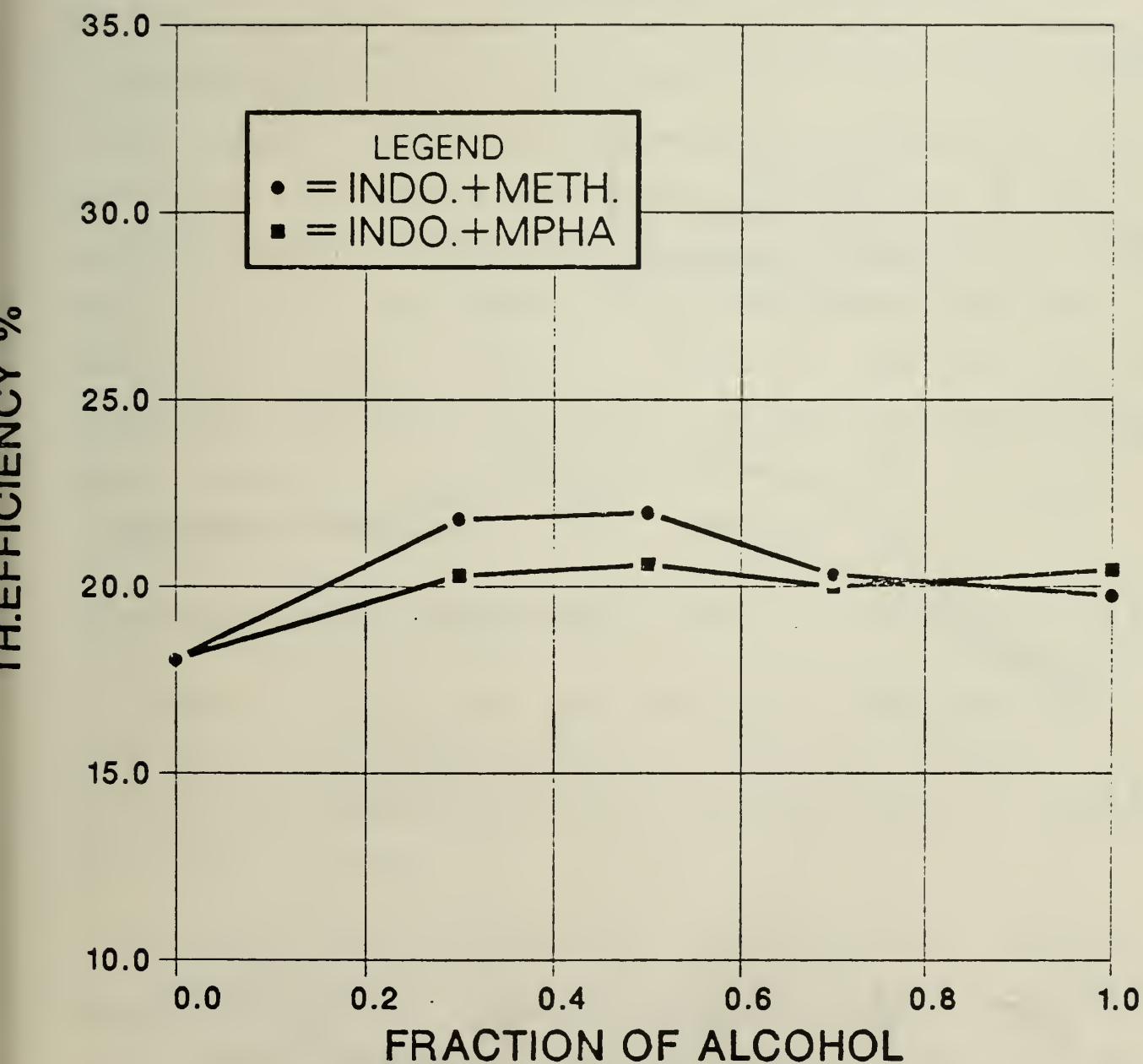


Fig. 5.27 thermal Efficiency at CR=7:1

THERMAL EFFICIENCY AT K.L.C.R.
INDO.-METH. & INDO.-MPHA BLENDS

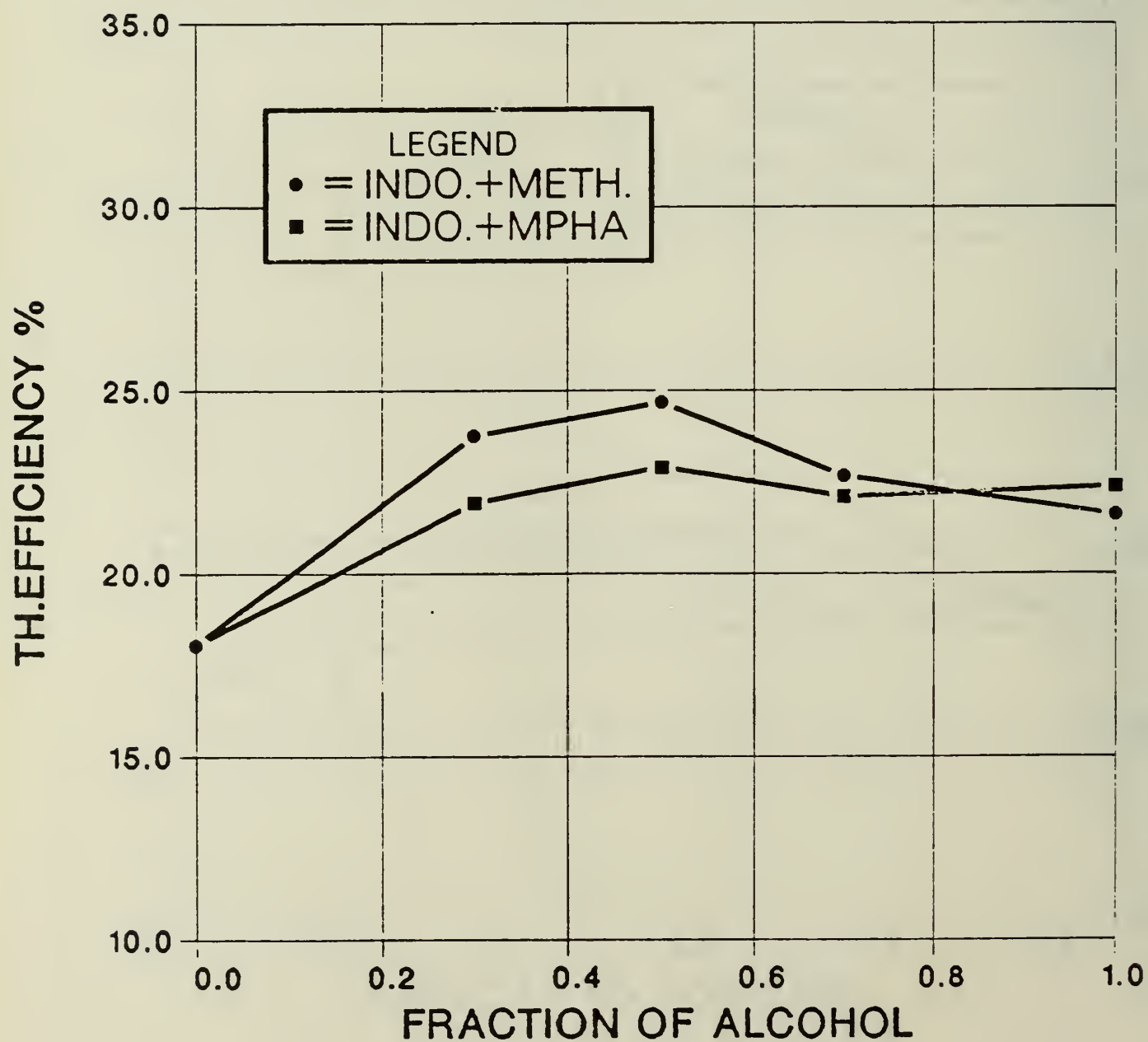


Fig. 5.28 Thermal Efficiency at CR=KLCR

required to explain the effect of indolene-MPHA on thermal efficiency.

Effect of Alcohol on Cycle-to Cycle Variation

Fig.5.29 shows the maximum standard deviation of the cylinder pressure in a cycle, as a function of fraction of alcohol, and also as a function of compression ratio. In general, as the alcohol concentration increases, the standard deviation increases. In other words, alcohol increases the cycle-to-cycle variation, although only marginally. The factors that affect cycle to cycle variation could be spark scatter, inhomogeneities in mixture temperature, ignition delay, and/or fluid motion in the vicinity of the spark plug. However, a more detailed analysis of the ignition delay, turbulence, and other parameters would have to be made before the exact causes for the increased cycle-to-cycle variation can be ascertained.

Effect of Alcohol on Flame Speed

Figures 5.30-5.40 show experimentally measured flame speed as a function of crank angle during the combustion process. No clear trend can be observed as to the effect of alcohol. This can be explained as follows:

The flame speed is function of the pressure and temperature of the unburned gases, turbulence, equivalence ratio, and the fuel type. Since the intake air was kept constant during this study, the mixture temperature varied according to fuel blend used. As the alcohol content increased, mixture temperature decreased, lowering

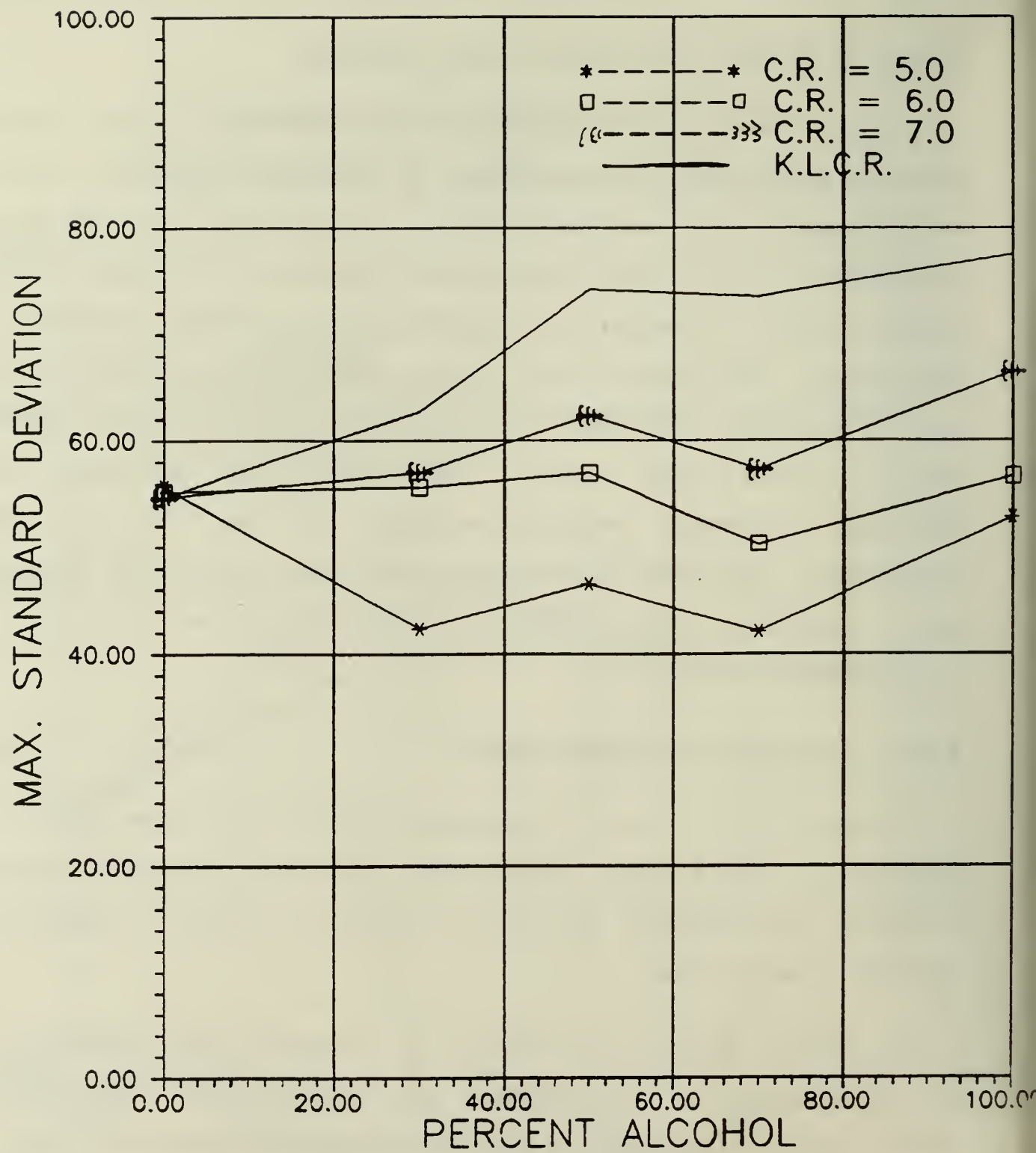
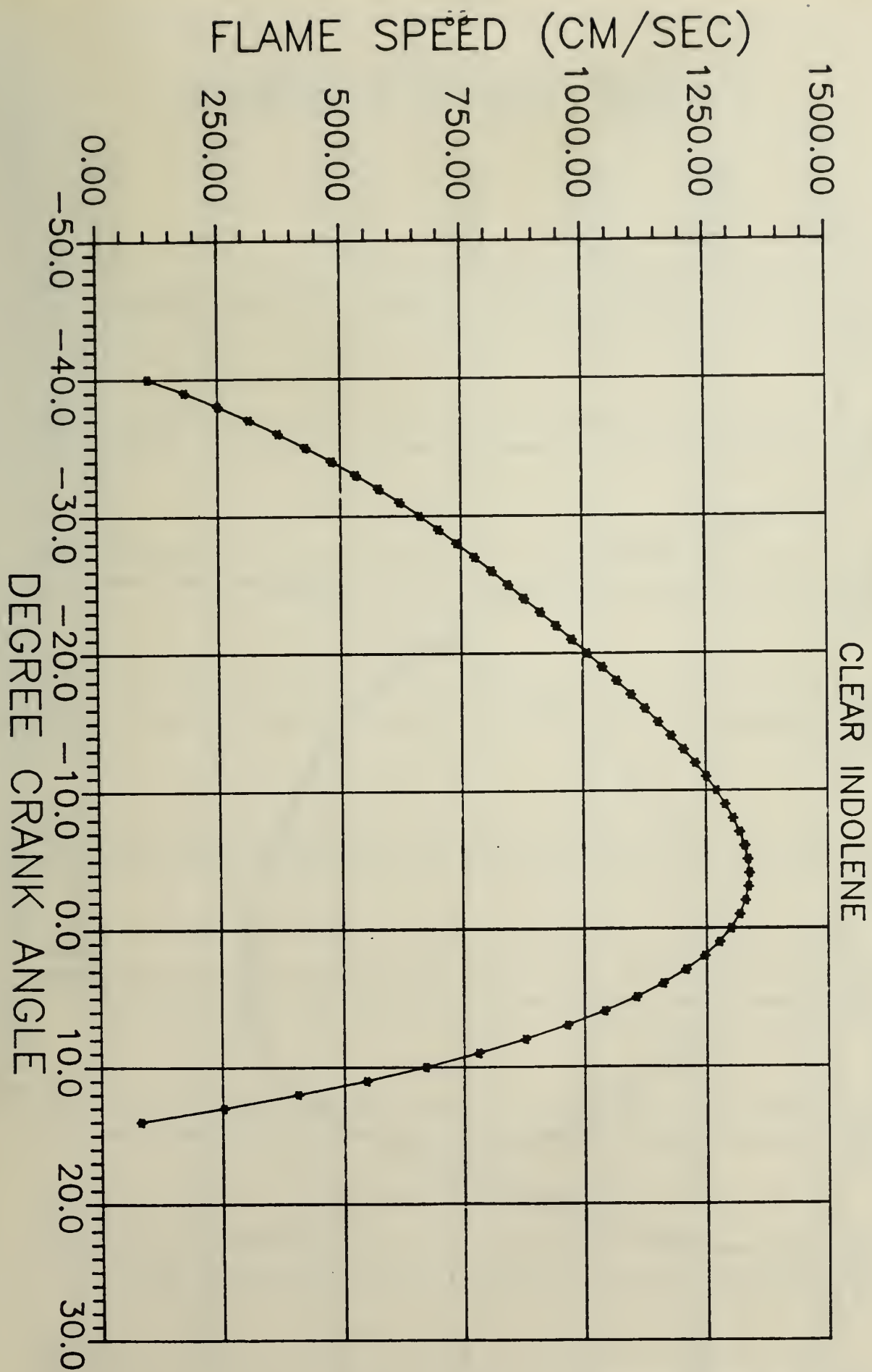


Fig. 5.29 Max. Standard Deviation



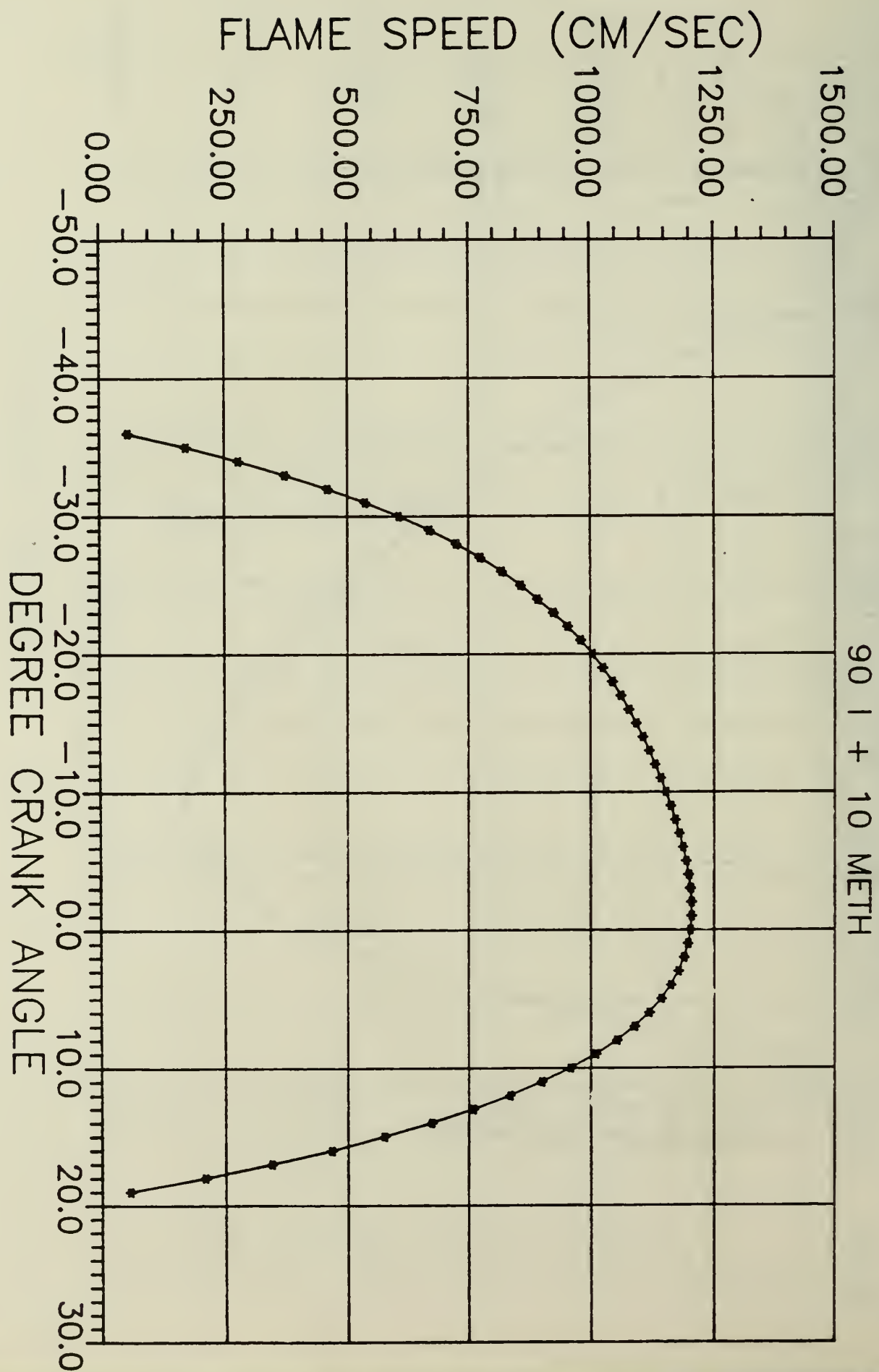
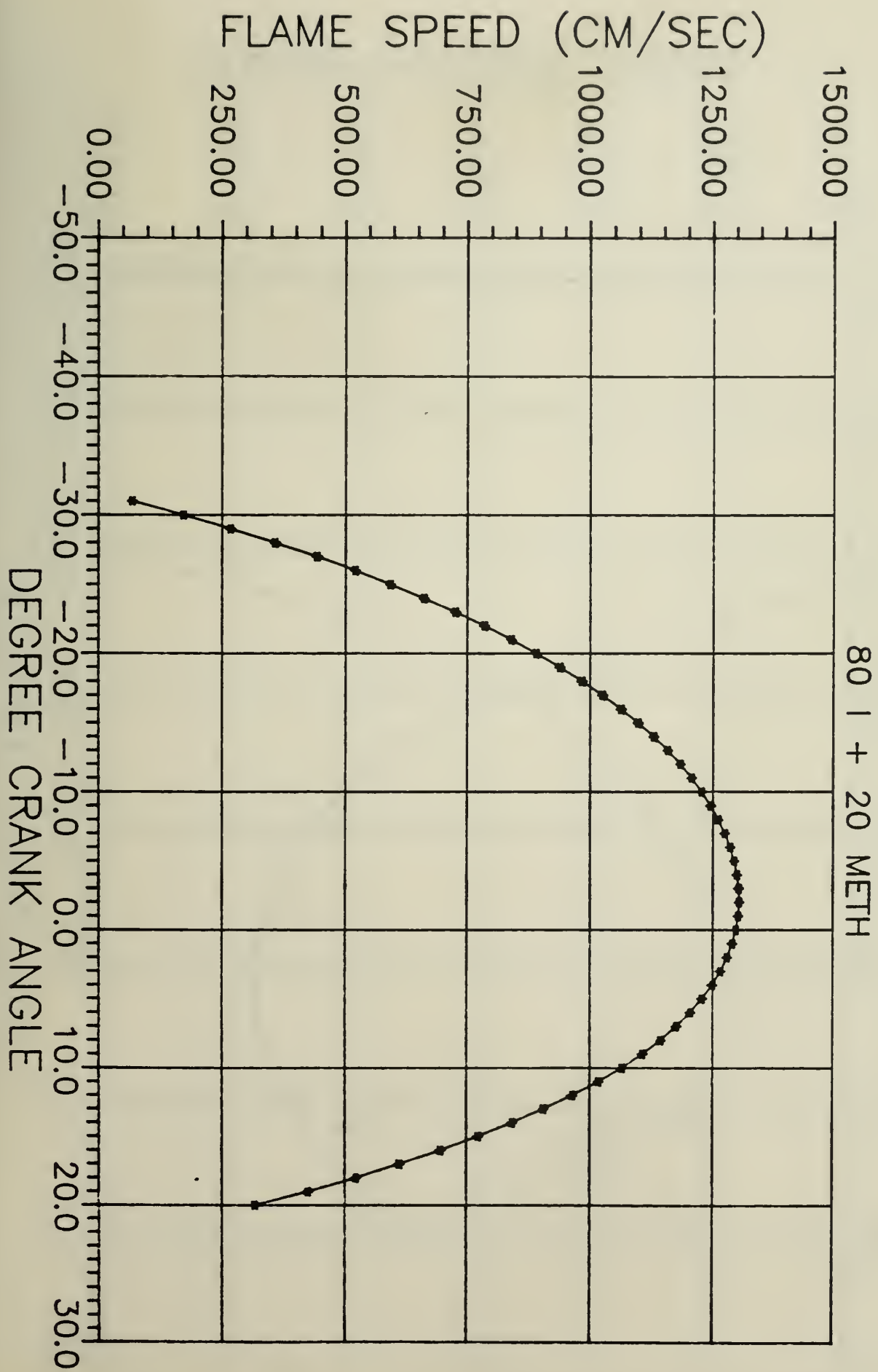


Fig. 5.31 Flame Speed of 10% Methanol Blend.



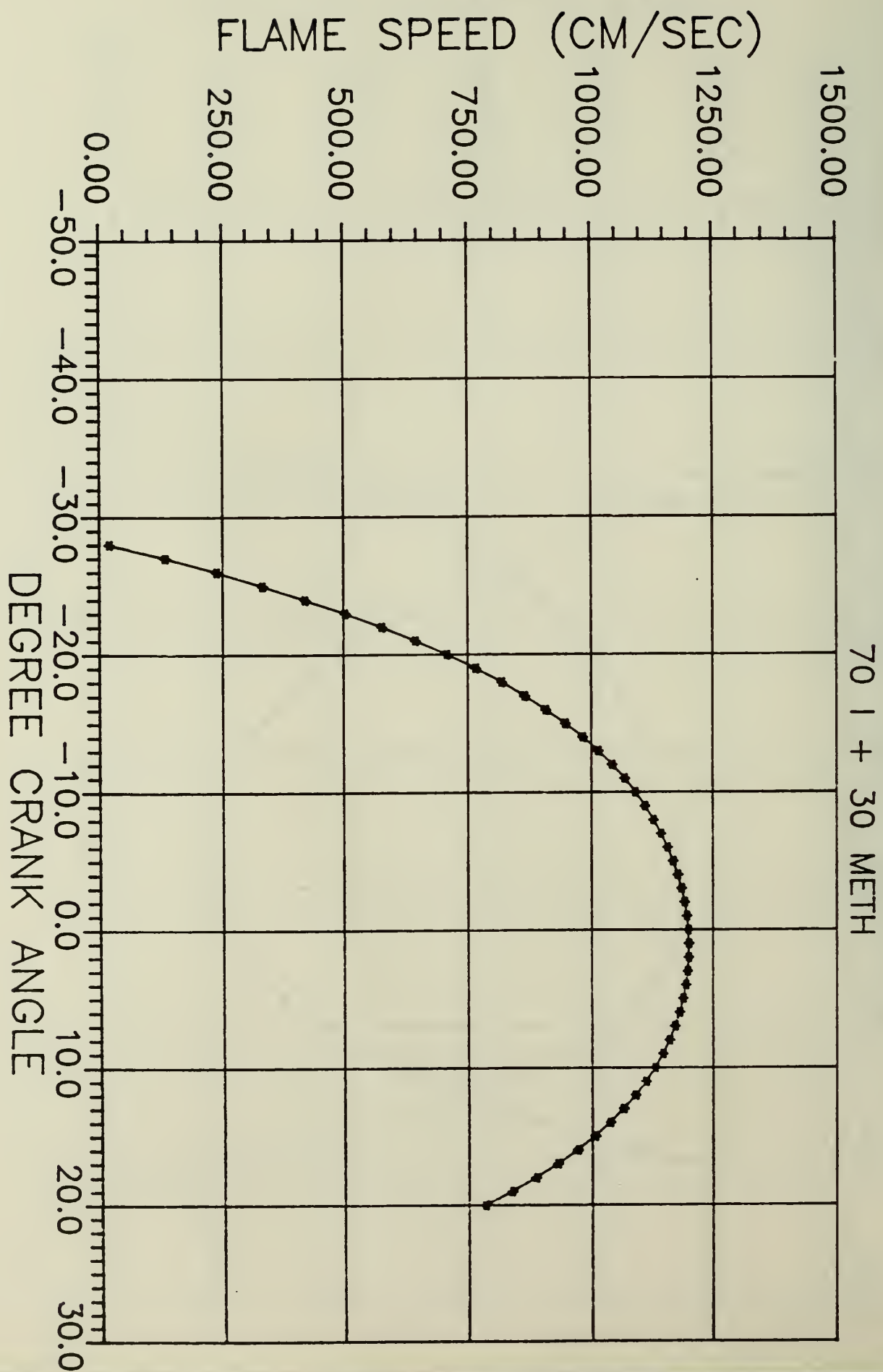
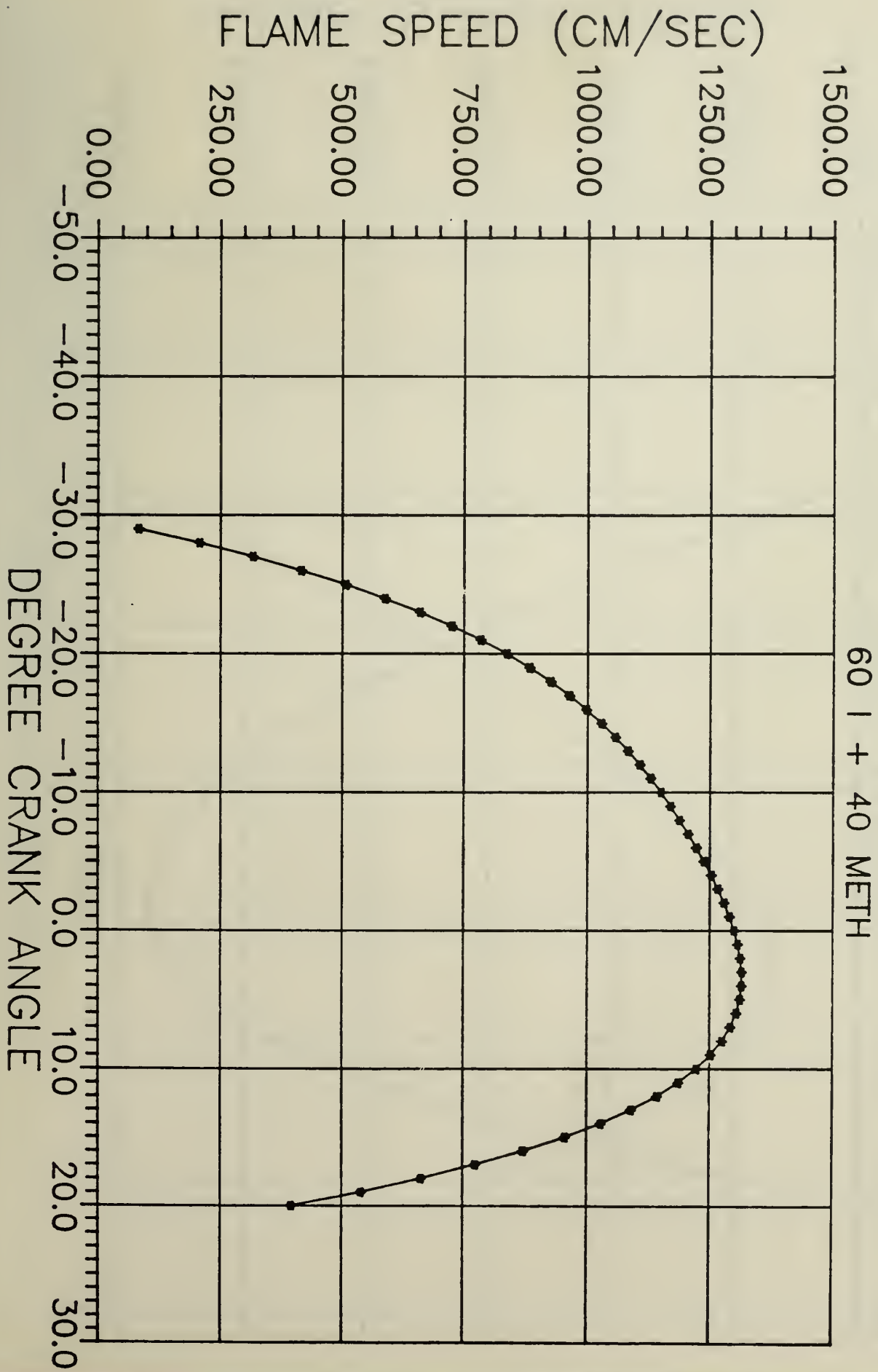


Fig. 5.33 Flame Speed of 30% Methanol Blend



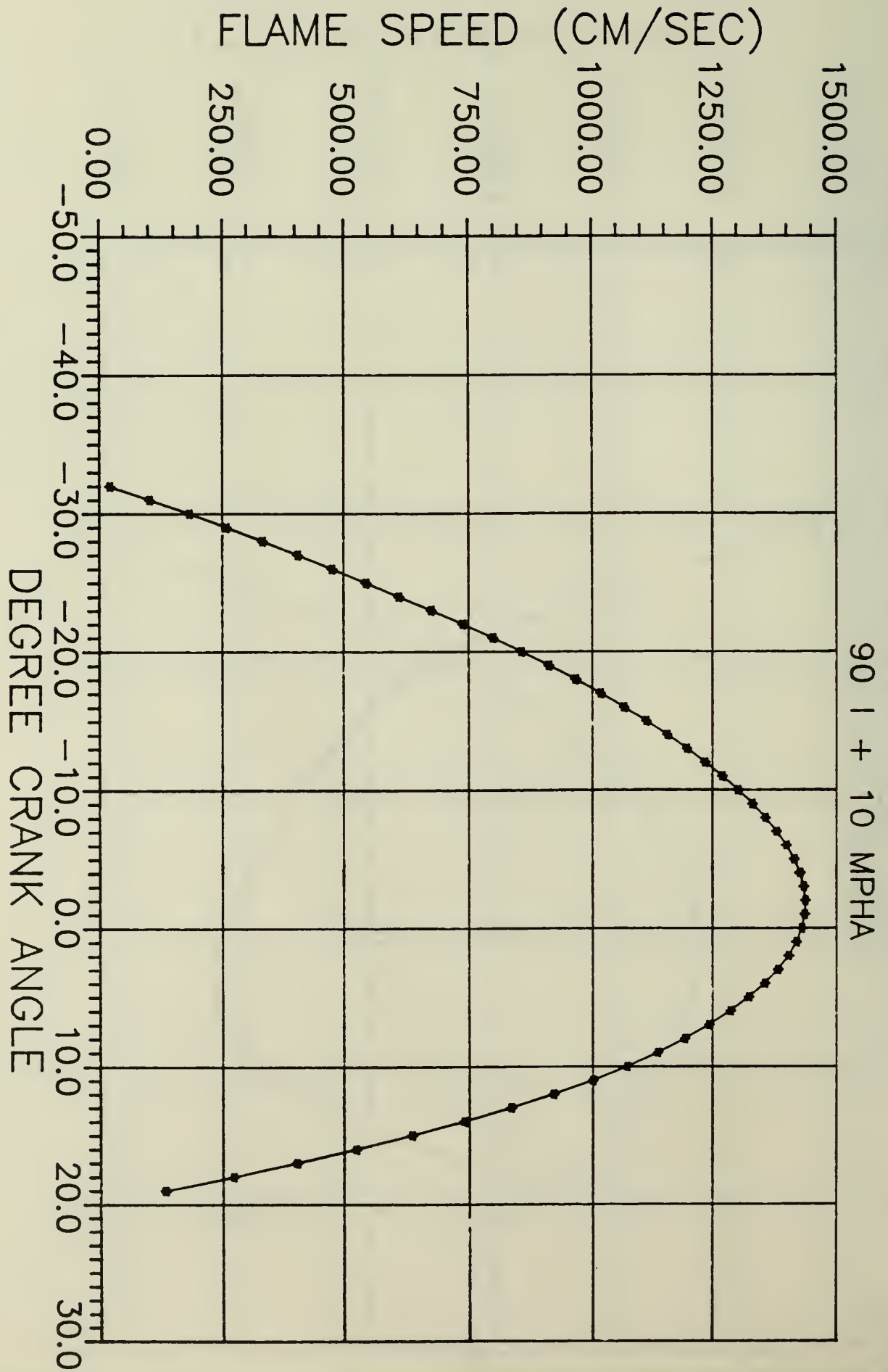
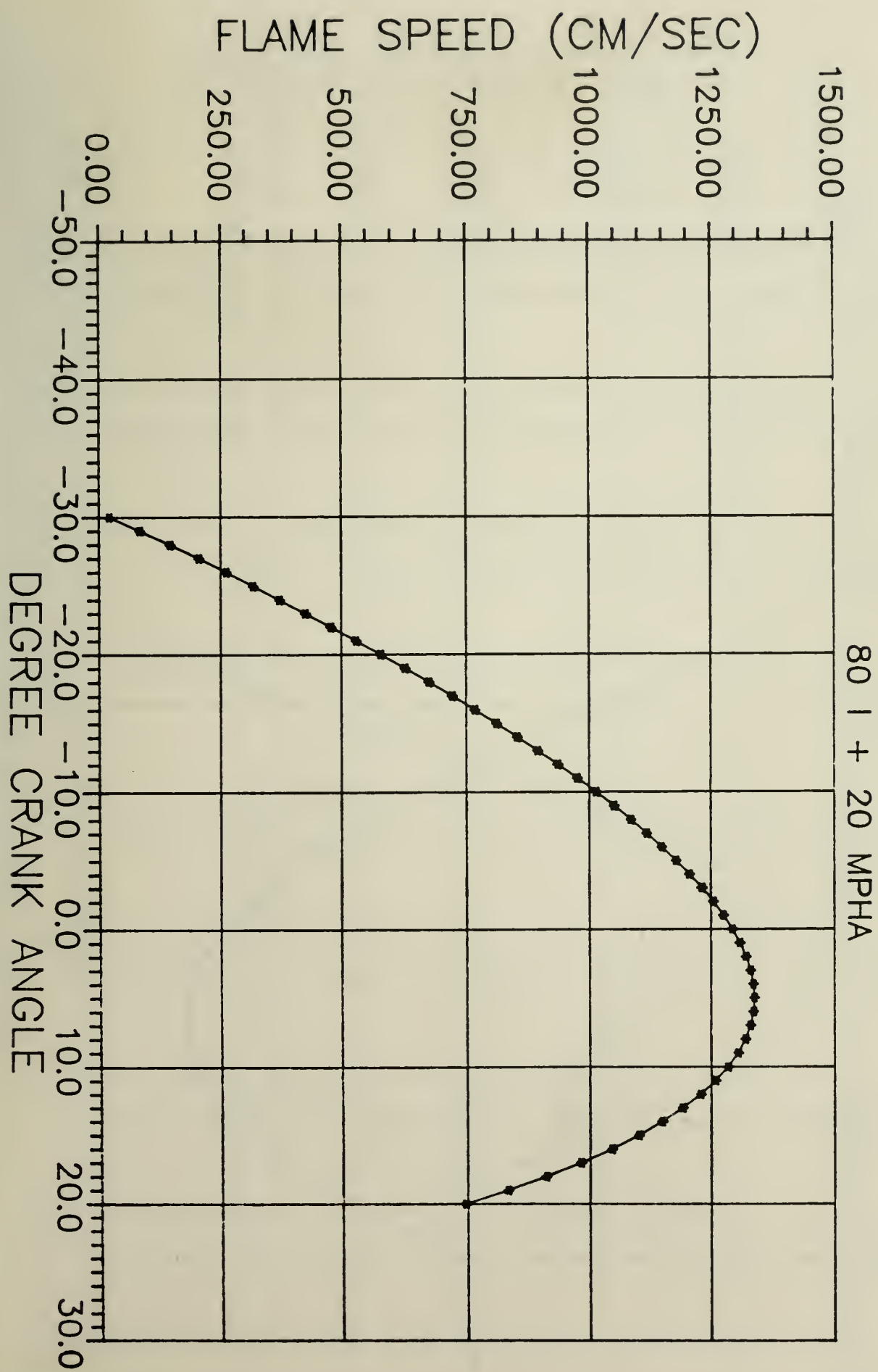


Fig. 5.35 Flame Speed of 10% MPHA Blend



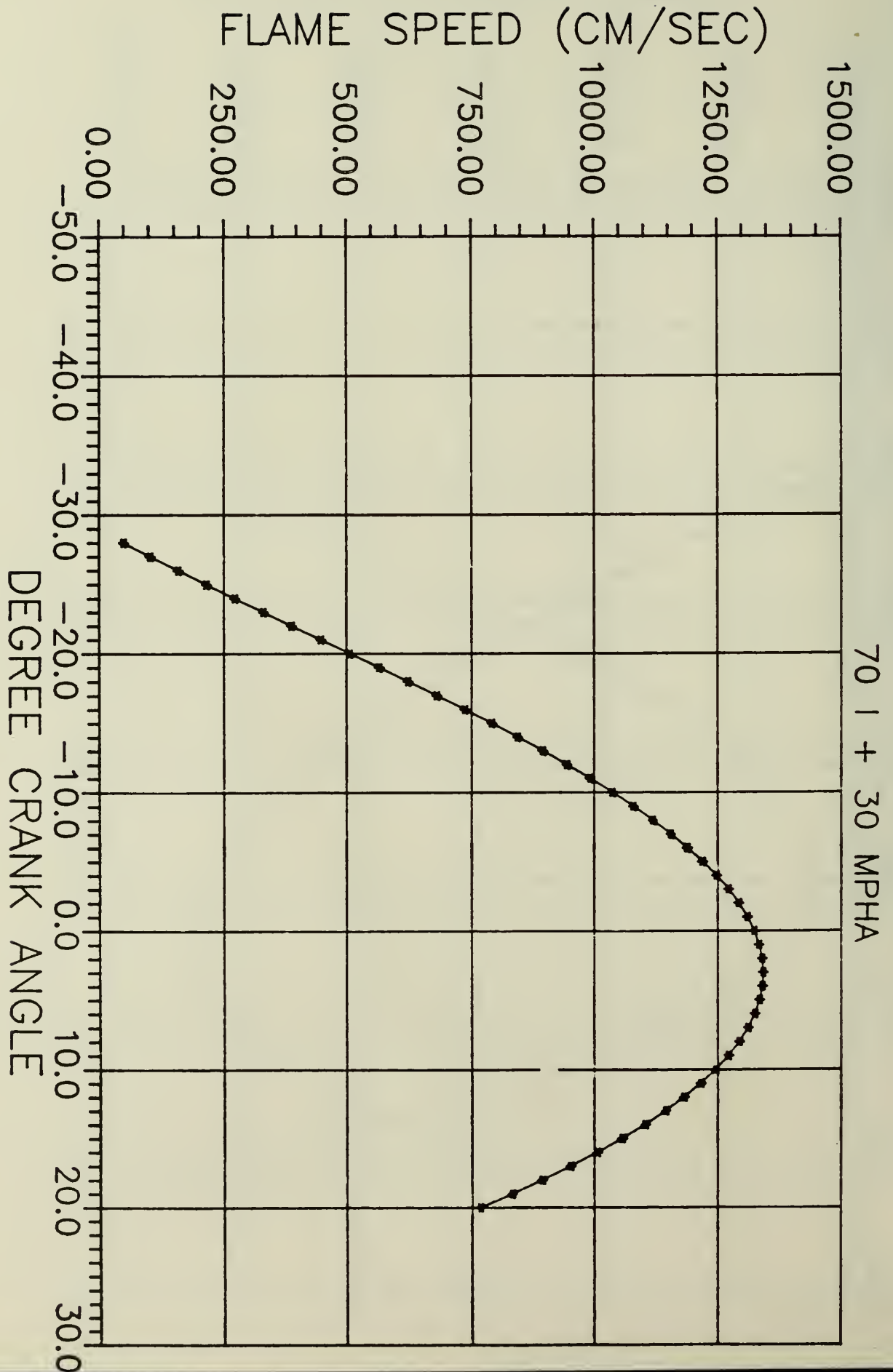


Fig. 5.37 Flame Speed of 30% MPHA Blend

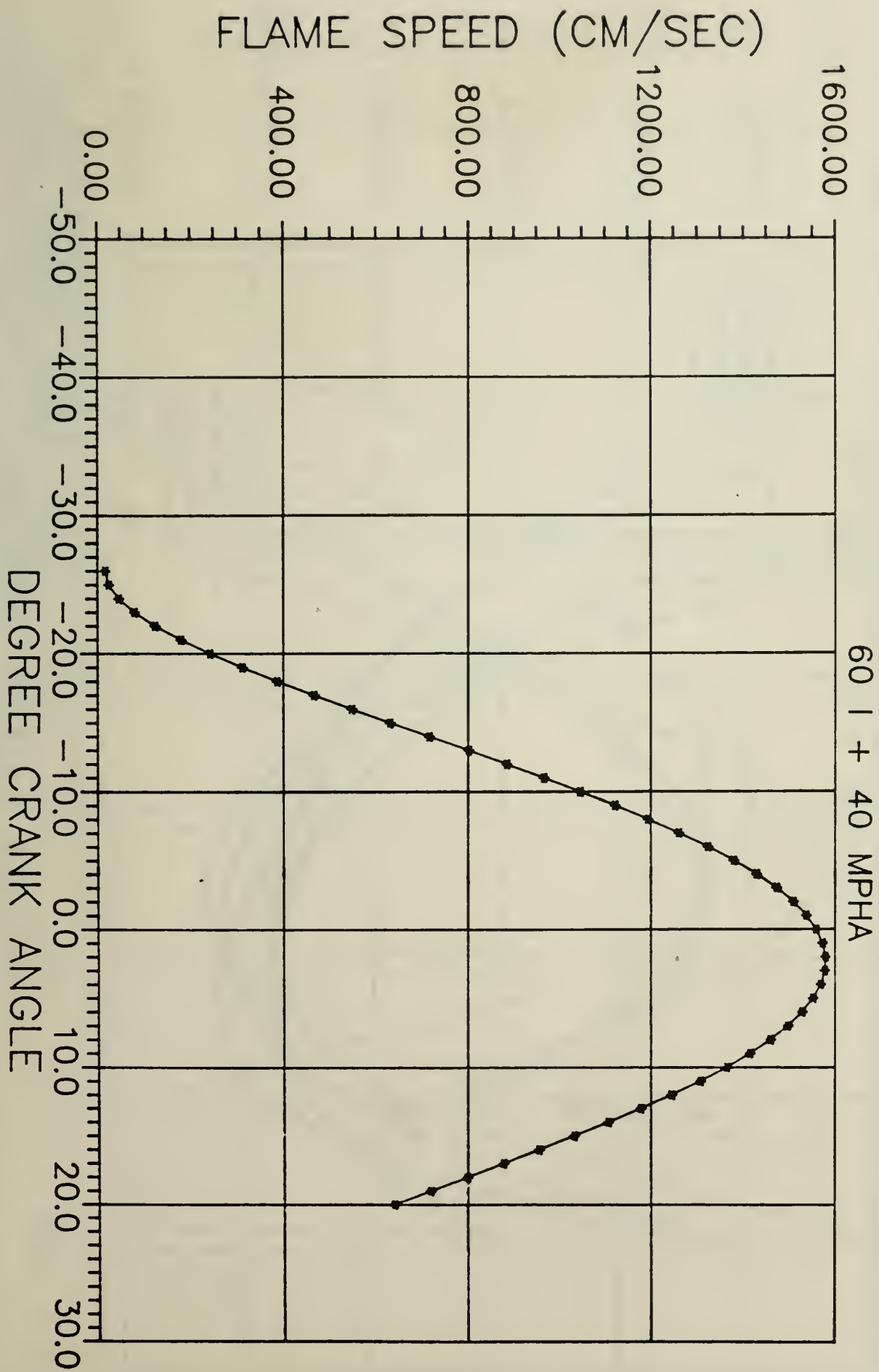


Fig. 5.38 Flame Speed of 40% MPHA Blend

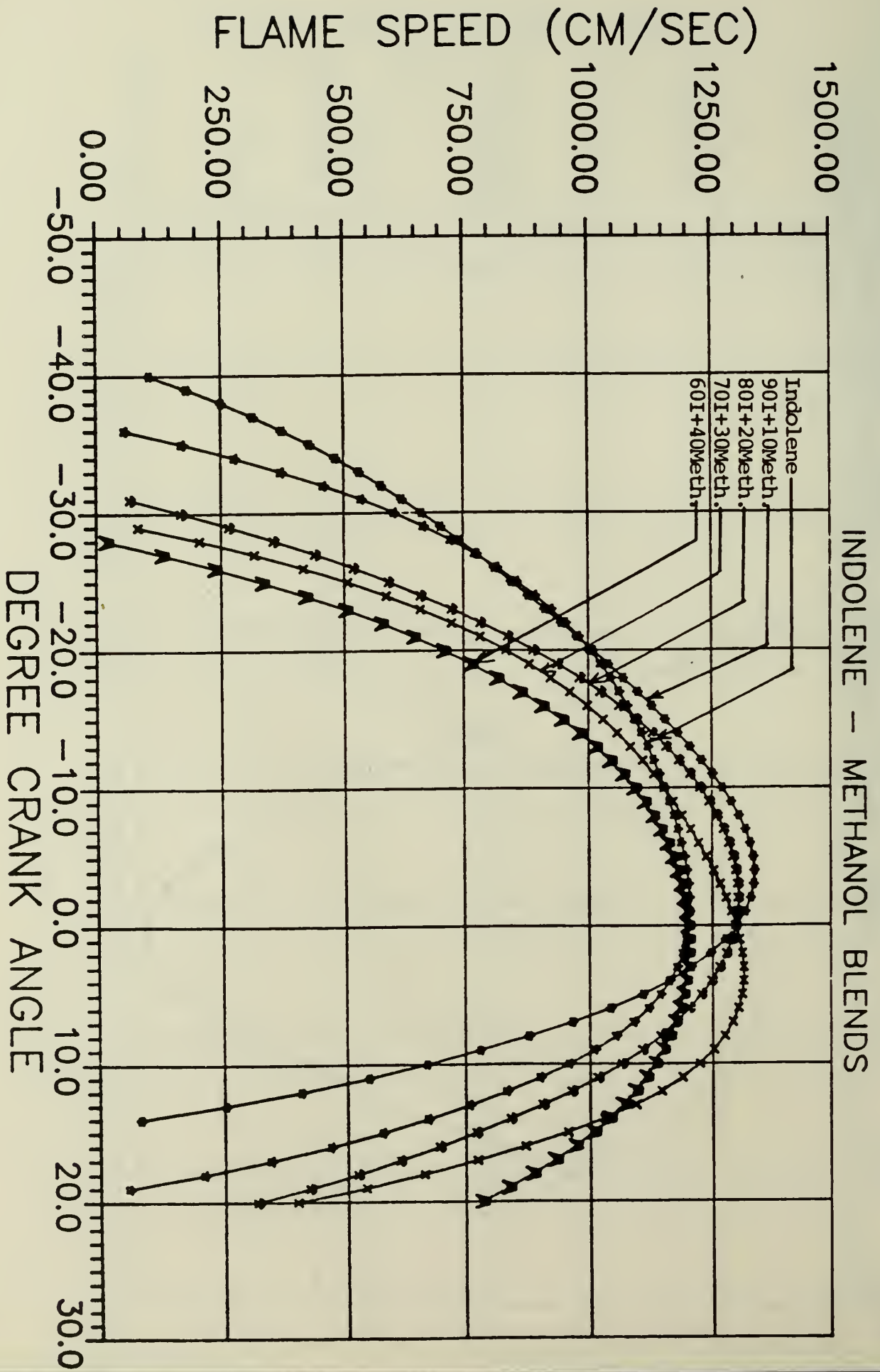
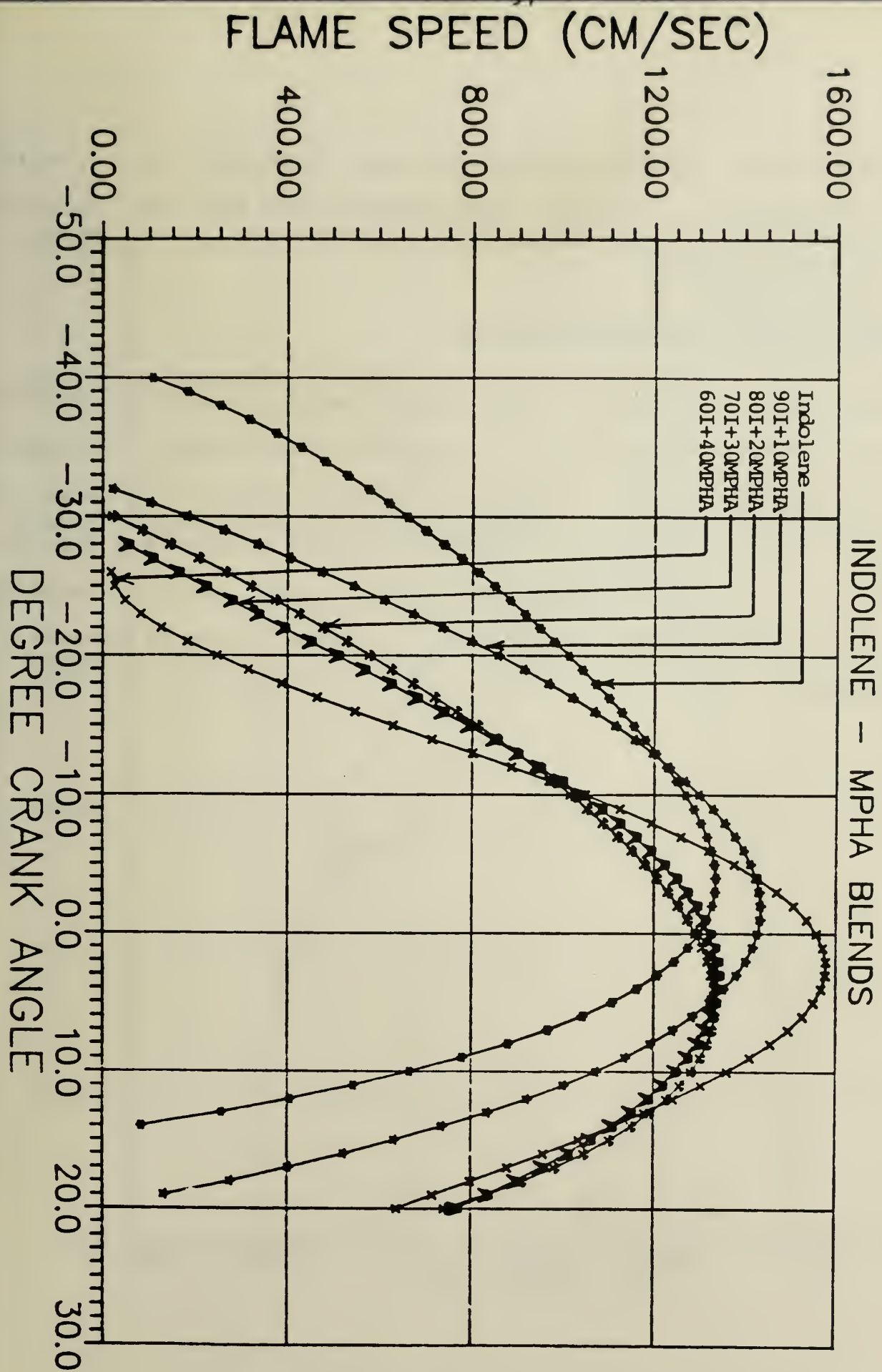


Fig. 5.39 Flame Speed of Indolene-Methanol Blends



the flame speed. Had the experiments been conducted at a constant mixture temperature, a higher flame speed would have been obtained with the alcohol blends.

Effect of Alcohol on Burning Velocity

As with the flame speed, the burning velocity is also affected by the mixture temperature. It is therefore inappropriate to compare the burning velocities of indolene-alcohol blends with those of indolene. It can be seen in Figures 5.41-5.44 that at any fraction of alcohol, indolene-MPHA blends have a higher burning velocity compared to indolene-methanol blends. However, the difference is only marginal.

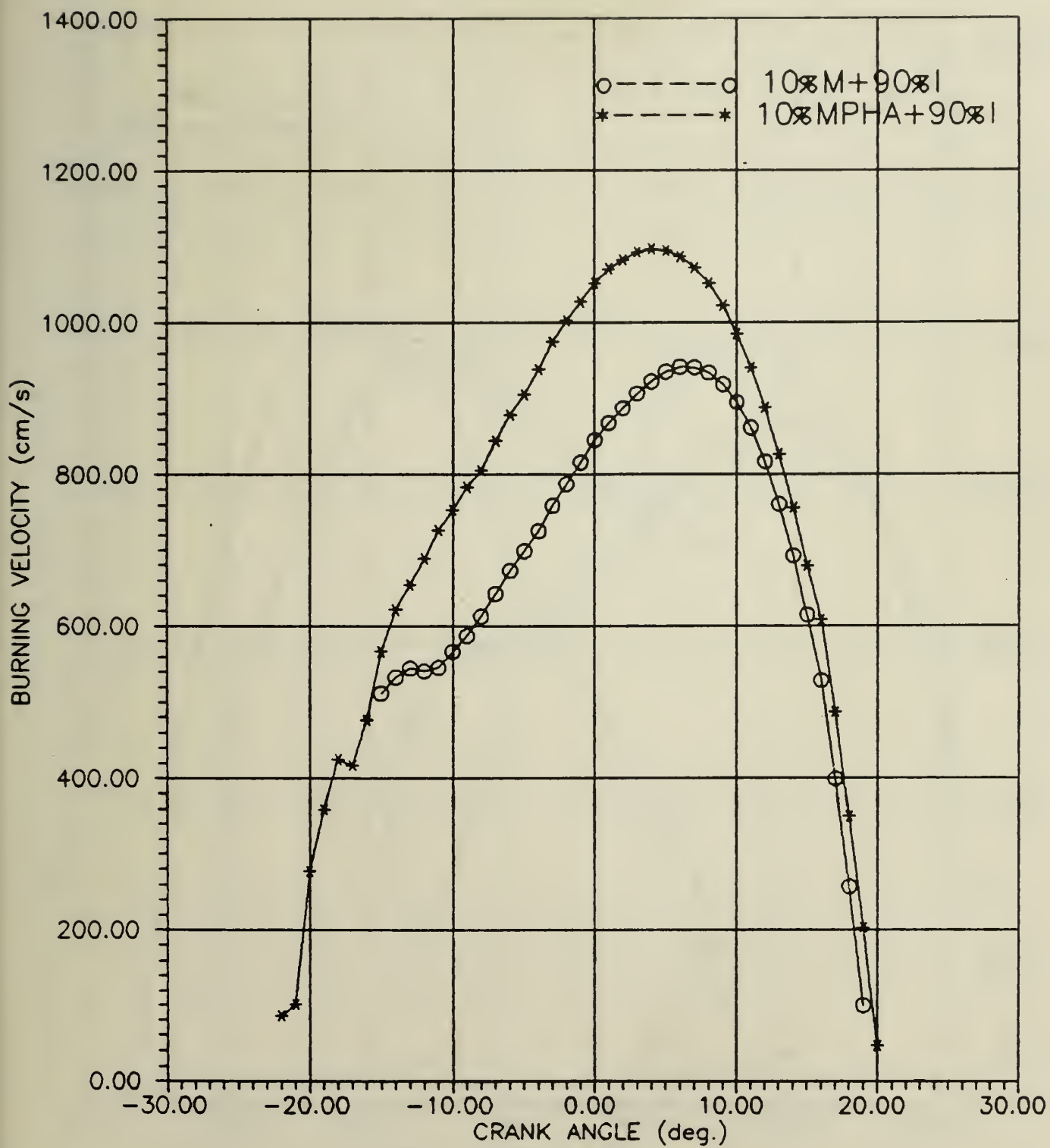


Fig. 5.41 Burning Velocity of 10% Blends.

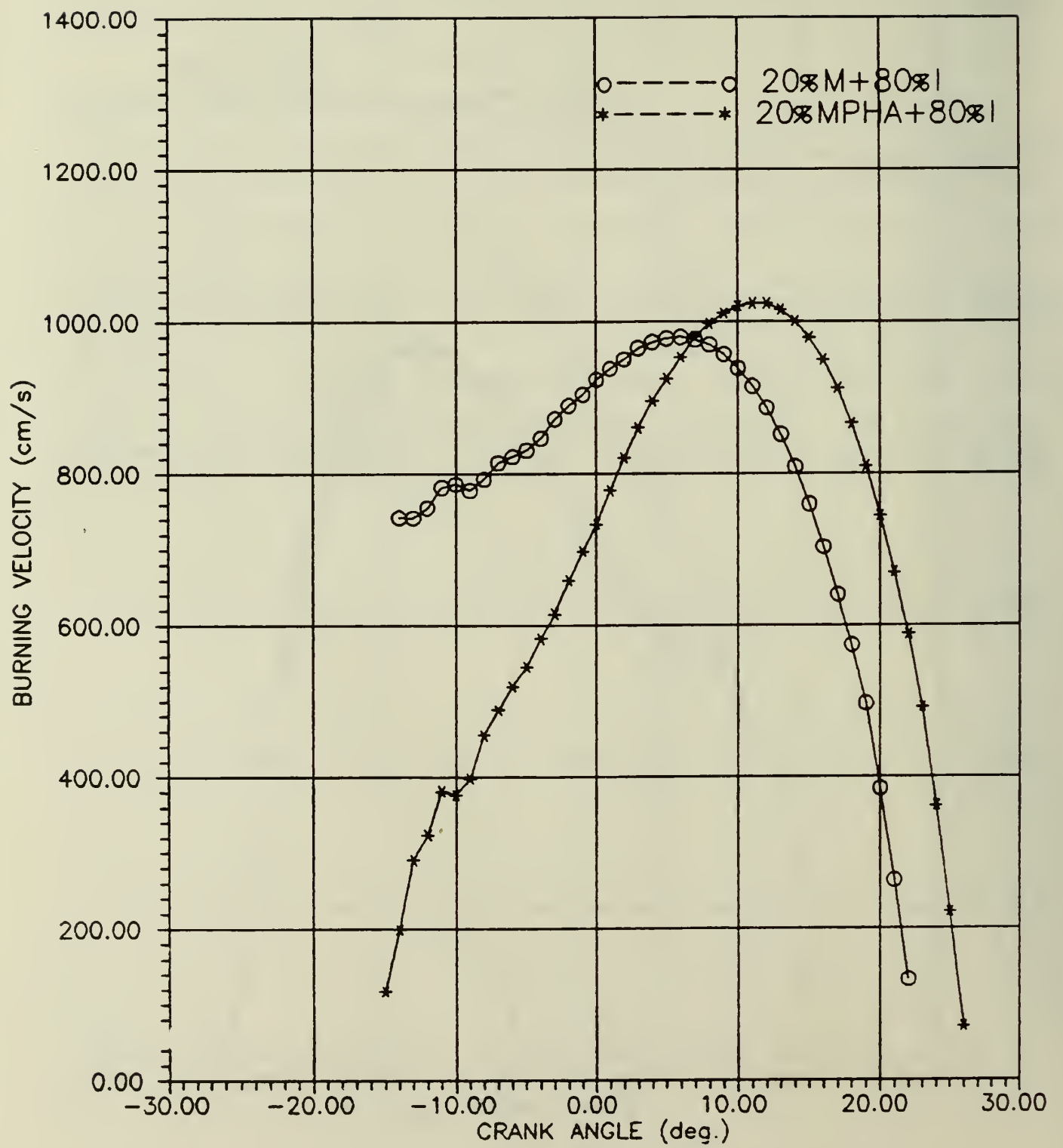


Fig. 5.42 Burning Velocity of 20% Blends

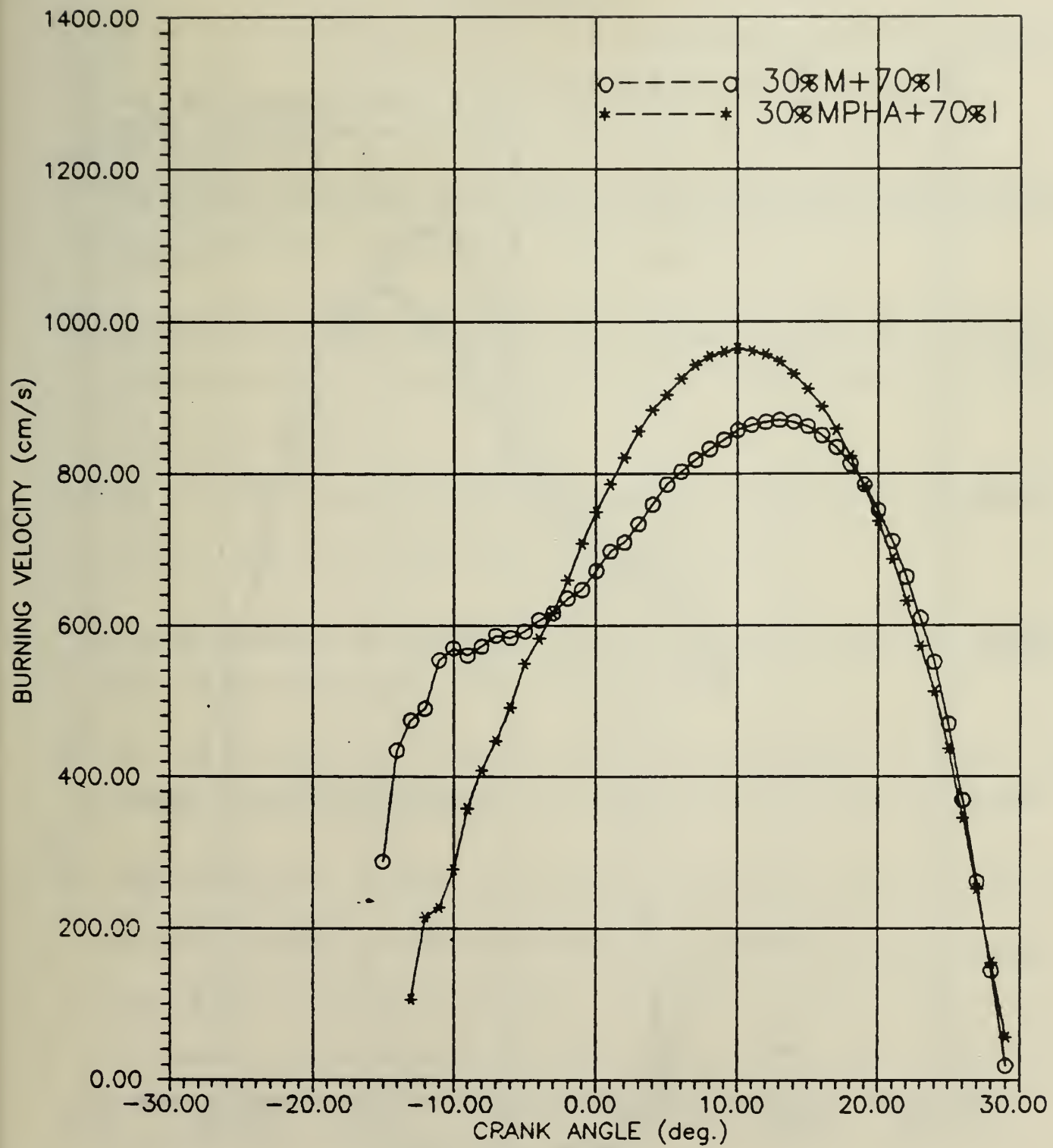


Fig. 5.43 Burning Velocity of 30% Blends

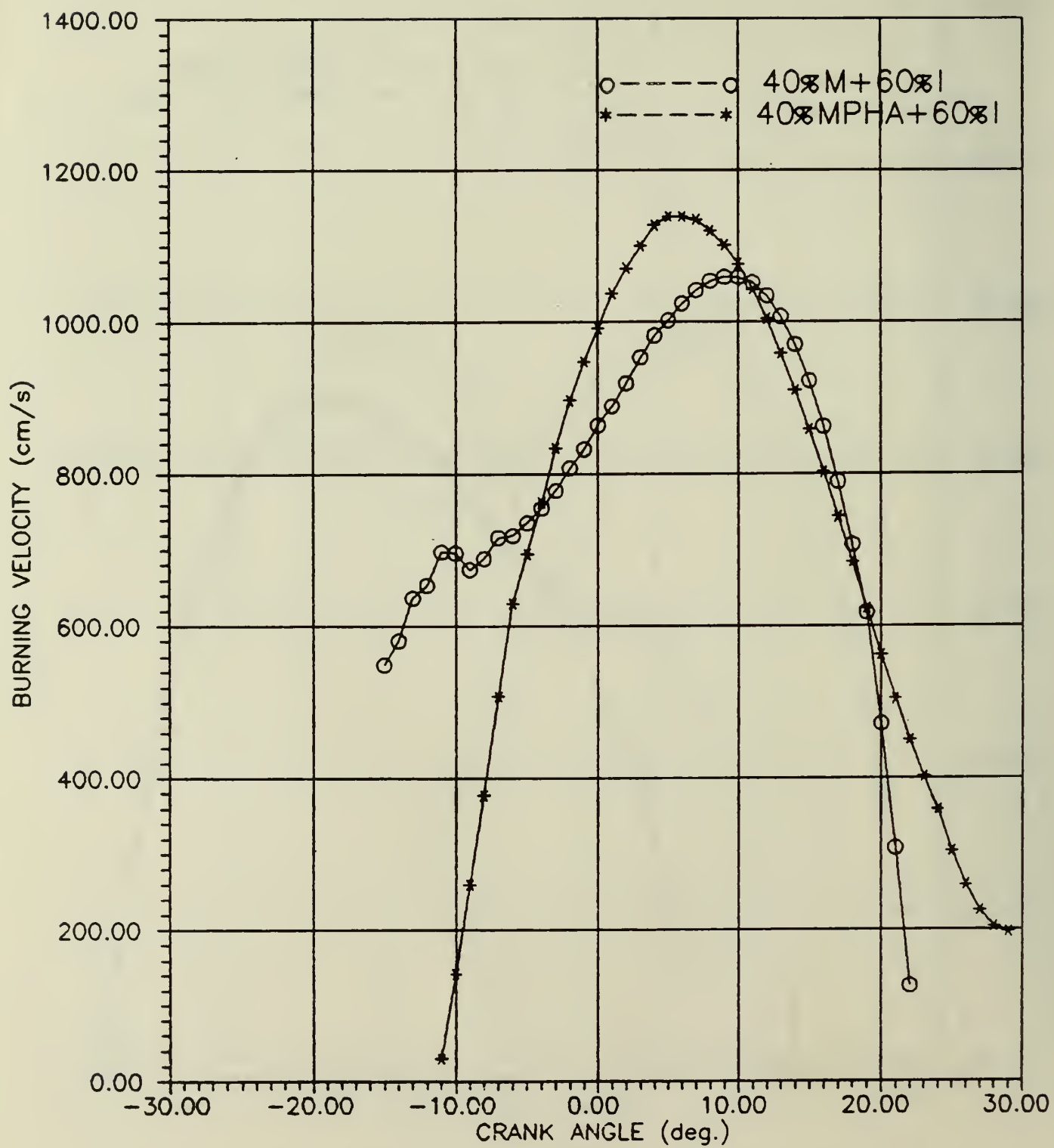


Fig. 5.44 Burning Velocity of 40% Blends.

CHAPTER VI

CONCLUSIONS

The following are the conclusions of the theoretical and experimental work.

1. The blend containing more than 20 percent alcohol did not meet 50 percent minimum distillation specification.
2. The heating value as measured with adiabatic bomb calorimeter, of indolene-MPHA is higher than that of indolene-methanol blend at all blend levels.
3. The water tolerance of indolene-MPHA blend is higher than indolene-methanol blend at all blend levels.
4. The API gravity of indolene-methanol is higher and the specific gravity is lower than indolene-MPHA blend.
5. The stoichiometric air-fuel ratio of indolene-MPHA is higher than indolene-methanol blend.
6. Adding methanol or MPHA to indolene causes the KLCR to increase. The KLCR of MPHA is greater than that of methanol.
7. MBT spark timing increases as the alcohol concentration increases. Indolene-methanol has a lower MBT spark advance than indolene-MPHA.
8. As methanol and MPHA concentration increases in the blend, power output, BSFC, and brake thermal efficiency increase. Indolene-

MPHA has the same power output, lower BSFC, and thermal efficiency compared to indolene-methanol blend.

9. The actual burning velocity of indolene-MPHA are slightly higher than indolene-methanol blends for the blends upto 40 percents. However there are mixed trends for flame speed.

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Supplementary Notes			
Abstract (Limit: 200 words) A study of the physical and chemical properties and performance of methanol plus higher alcohol (MPHA)-indolene has been completed. The fuel properties investigated are distillation characteristics, heat of combustion, flash point, specific gravity, water tolerance and stoichiometry. The performance parameters investigated are MBT spark advance, knock limited compression ratio, brake power, brake mean effective pressure, brake thermal efficiency and brake specific fuel consumption. The fuel properties of fifteen indolene-alcohol blends were measured at different alcohol con- centration in indolene-alcohol blends.			
Document Analysis a. Descriptors methyl alcohol, alcohols gasoline, blends, spark ignition engines, automotive fuels, physical properties, performance tests.			
b. Identifiers/Open-Ended Terms			
c. COSATI Field/Group			
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